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| IDENTIFIERS | *Millwrights; *Turbines |

ABSTRACT

This packet, part of the instructional materials for the Oregon apprenticeship program for millwright training, contains five modules covering turbines. The modules provide information on the following topics: types, components, and auxiliaries of steam turbines; operation and maintenance of steam turbines; and gas turbines. Each module consists of a goal, performance indicators, student study guide, vocabulary, introduction, information sheets illustrated with line drawings and photographs, an assignment sheet, a job sheet, a self-assessment test with answers, a post-assessment test with answers for the instructor, and a list of supplementary references. (Copies of supplementary references, which are sections of lectures from a correspondence course published by the Southern Alberta Institute of Technology, are included in the packets.) (KC)

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ED2547722

APPRENTICESHIP

# MILLWRIGHT

RELATED
TRAINING MODULES

8.1-8.5 TURBINES

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APPRENTICESHIP

MILLWRIGHT
RELATED TRAINING MODULES

SAFETY

1.1 General Safety
1.2 Hand Tool Safety
1.3 Power Tool Safety
1.4 Fire Safety
1.5 Hygiene Safety
1.6 Safety and Electricity
1.7 Fire Types and Prevention
1.8 Machine Safeguarding (includes OSHA Handbook)

ELECTRICITY/ELECTRONICS

2.1 Basics of Energy
2.2 Atomic Theory
2.3 Electrical Conduction
2.4 Basics of Direct Current
2.5 Introduction to Circuits
2.6 Reading Scales
2.7 Using a V.O.M.
2.8 OHM'S Law
2.9 Power and Watt's Law
2.10 Kirchoff's Current Law
2.11 Kirchoff's Voltage Law
2.12 Series Resistive Circuits
2.13 Parallel Resistive Circuits
2.14 Series - Parallel Resistive Circuits
2.15 Switches and Relays
2.16 Basics of Alternating Currents
2.17 Magnetism

COMPUTERS

3.1 Digital Language
3.2 Digital Logic
3.3 Computer Overview
3.4 Computer Software

TOOLS

4.1 Boring and Drilling Tools
4.2 Cutting Tools, Files and Abrasives
4.3 Holding and Fastening Tools
4.4 Fastening Devices
4.5 Basic Science - Simple Mechanics
4.6 Fasteners

## DRAFTING

5.1 Types of Drawing and Views
5.2 Sketching
5.3 Blueprint Reading/Working Drawings
5.4 Working Drawings for Machines and Welding
5.5 Machine and Welding Symbols
5.6 Blueprint Reading, Drafting: Basic Print Reading
5.7 Blueprint Reading, Drafting: Basic Print Reading
5.8 Blueprint Reading, Drafting: Basic Print Reading
5.9 Blueprint Reading, Drafting: Basic Print Reading
5.10 Blueprint Reading, Drafting: Basic Print Reading
5.11 Blueprint Reading, Drafting: Basic Print Reading
5.12 Blueprint Reading, Drafting: Basic Print Reading
5.13 Blueprint Reading, Drafting: Basic Print Reading
5.14 Drafting, Machine Features
5.15 Drafting, Measurement
5.16 Drafting, Visualization

## HUMAN RELATIONS

6.1 Communications Skills
6.2 Feedback
6.3 Individual Strengths
6.4 Interpersonal Conflicts
6.5 Group Problem Solving
6.6 Goal-setting and Decision-making
6.7 Worksite Visits
6.8 Resumes
6.9 Interviews
6.10 Expectation
6.11 Wider Influences and Responsibilities
6.12 Personal Finance

## BOILERS

7.1 Boilers - Fire Tube Types
7.2 Boilers - Watertube Types
7.3 Boilers - Construction
7.4 Boilers - Fittings
7.5 Boilers - Operation
7.6 Boilers - Cleaning
7.7 Boilers - Heat Recovery Systems
7.8 Boilers - Instruments and Controls
7.9 Boilers - Piping and Steam Traps

## TURBINES

8.1 Steam Turbines - Types
8.2 Steam Turbines - Components
8.3 Steam Turbines - Auxillaries
8.4 Steam Turbines - Operation and Maintenance
8.5 Gas Turbines

PUMPS

9.1 Pumps - Types and Classification
9.2 Pumps - Applications
9.3 Pumps - Construction
9.4 Pumps - Calculating Heat and Flow
9.5 Pumps - Operation
9.6 Pumps - Monitoring and Troubleshooting
9.7 Pumps - Maintenance

COMBUSTION

10.1 Combustion - Process
10.2 Combustion - Types of Fuel
10.3 Combustion - Air and Fuel Gases
10.4 Combustion - Heat Transfer
10.5 Combustion - Wood

GENERATORS

11.1 Generators - Types and Construction
11.2 Generators - Operation

FEEDWATER

12.1 Feedwater - Types and Equipment
12.2 Feedwater - Water Treatments
12.3 Feedwater - Testing

AIR COMPRESSORS

13.1 Air Compressors - Types
13.2 Air Compressors - Operation and Maintenance

STEAM

14.1 Steam - Formation and Evaporation
14.2 Steam - Types
14.3 Steam - Transport
14.4 Steam - Purification

MISCELLANEOUS

15.1 Installation - Foundations
15.2 Installation - Alignment
15.3 Circuit Protection
15.4 Transformers
15.5 Trade Terms

TRADE MATH

16.1 Linear - Measure
16.2 Whole Numbers
16.3 Additional and Subtraction of Common Fraction and Mixed Numbers
16.4 Multiplication and Division of Common Fractions and Whole and Mixed Numbers

16.5 Compound Numbers
16.6 Percent
16.7 Ratio and Proportion
16.8 Perimeters, Areas and Volumes
16.9 Circumference and Wide Area of Circles
16.10 Area of Plane, Figures and Volumes of Solid Figures
16.11 Metrics

HYDRAULICS

17.1 Hydraulics - Lever
17.2 Hydraulics - Transmission of Force
17.3 Hydraulics - Symbols
17.4 Hydraulics - Basic Systems
17.5 Hydraulics - Pumps
17.6 Hydraulics - Pressure Relief Valve
17.7 Hydraulics - Reservoirs
17.8 Hydraulics - Directional Control Valve
17.9 Hydraulics - Cylinders
17.10 Hydraulics - Forces, Area, Pressure
17.11 Hydraulics - Conductors and Connectors
17.12 Hydraulics - Troubleshooting
17.13 Hydraulics - Maintenance

METALLURGY

18.1 Included are ILS packets:
W 3010
W 3011-1
W 3011-2
MS 9001 (1-3-4-8-9-6-7-5-2-9)
MS 9200, 9201

POWER DRIVES

19.1 101. A-B-C-D-E
102. C-D-E
103. B-C-D-E
104. A-C-E-F-G-H-I-J
107. A
108. A

WELDING

20.1 602. A-B-C-D-G-I-L-M
603. A-B-F-G-I
W. 3011-1 refer to Mettallurgy 18.1
WE. MA-18

MILLWRIGHT
SUPPLEMENTARY REFERENCE DIRECTORY

Note: All reference packets are numbered on the upper right-hand corner of the respective cover page.

| Supplementary Packet # | Description | Related Training Module | |
|---|---|---|---|
| 1.8 | Concepts & Techniques of Machine Safeguarding, U.S.D.L., O.S.H.A. | 1.8 | Machine Safeguarding |
| 12.1 | Correspondence Course, Lecture 1, Sec. 2, Steam Generators, Types of Boilers I, S.A.I.T., Calgary, Alberta, Canada | 7.1 | Boilers, Fire Tube Type |
| 12.2 | Correspondence Course, Lecture 2, Sec. 2, Steam Generators, Types of Boikers II, S.A.I.T., Calgary, Alberta, Canada | 7.2 | Boilers, Water Tube Type |
| 12.3 | Correspondence Course, Lecture 2, Sec. 2, Steam Generators, Boiler Construction & Erection, S.A.I.T., Calgary, Alberta, Canada | 7.3 | Boilers, Construction |
| 12.4 | Correspondence Course, Lecture 4, Sec. 2, Steam Generators, Boiler Fittings II, S.A.I.T., Calgary, Alberta, Canada | 7.4 | Boilers, Fittings |
| 12.4 | Correspondence Course, Lecture 4, Sec. 2, Steam Generators, Boiler Fitting I, S.A.I.T., Calgary, Alberta, Canada | 7.4 | Boilers, Fittings |
| 12.5 | Correspondence Course, Lecture 10, Sec. 2, Steam Generation, Boiler Operation, Maintenance, Inspection, S.A.I.T., Calgary, Alberta, Canada | 7.5 | Boilers, Operation |
| 12.7 | Correspondence Course, Lecture 3, Sec. 2, Steam Generation, Boiler Details, S.A.I.T., Calgary, Alberta, Canada | 7.7 | Boilers Heat Recovery Systems |
| | | PUMPS | |
| 13.1 | Correspondence Course, Lecture 9, Sec. 2, Steam Generator, Power Plant Pumps, S.A.I.T., Calgary, Alberta, Canada | 9.1 | Types & Classifications |
| 13.2 | | 9.2 | Applications |
| 13.4 | | 9.4 | Calculating Heat & Flow |
| 13.6 | | 9.6 | Monitoring & Troubleshooting |
| 13.7 | | 9.7 | Maintenance |
| 13.3 | Correspondence Course, Lecture 6, Sec. 3, Steam Generators, Pumps, S.A.I.T., Calgary, Alberta, Canada | 9.3 | Construction |
| 13.5 | | 9.5 | Operation |

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| Supplementary Packet # | Description | Related Training Module |
|---|---|---|
| 14.3<br>12.8 | Correspondence Course, Lecture 6, Sec. 3, Steam Generators, Steam Generator Controls, S.A.I.T., Calgary, Alberta, Canada | 14.3 Steam Transport<br>7.8 Boilers, Instruments & Controls |
| 14.4 | Correspondence Course, Lecture 11, Sec. 2, Steam Generators, Piping II, S.A.I.T., Calgary, Alberta, Canada | 14.4 Steam Purification |
| 15.1 | Correspondence Course, Lecture 1, Sec. 4, Prime Movers, & Auxiliaries, Steam Turbines, S.A.I.T., Calgary, Alberta, Canada | 8.1 Steam Turbines, Types |
| 15.2 | Correspondence Course, Lecture 4, Sec. 3, Prime Movers, Steam Turbines I, S.A.I.T., Calgary, Alberta, Canada | 8.2 Steam Turbines, Components |
| 15.3 | Correspondence Course, Lecture 2, Sec. 4, Prime Movers & Auxiliaries, Steam Turbine Auxiliaries, S.A.I.T., Calgary, Alberta, Canada | 8.3 Steam Turbines, Auxiliaries |
| 15.4 | Correspondence Course, Lecture 6, Sec. 3, Prime Movers, Steam Turbine Operation & Maintenance, S.A.I.T., Calgary, Alberta, Canada | 8.4 Steam Turbines, Operation & Maintenance |
| 15.5 | Correspondence Course, Lecture 8, Sec. 3, Prime Movers, Gas Turbines, S.A.I.T., Calgary, Alberta, Canada | 8.5 Gas Turbines |
| 16.2 | Boilers Fired with Wood & Bark Residues, D.D. Junge, F.R.L., O.S.U., 1975 | 10.2 Combustion Types of Fuel |
| 16.2 | Correspondence Course, Lecture 5, Sec. 2, Steam Generators, Fuel Combustion, S.A.I.T., Calgary, Alberta, Canada | 10.2 Combustion Types of Fuel |
| 16.3 | Correspondence Course, Lecture 5, Sec. 2, Plant Services, Fuel & Combustion, S.A.I.T., Calgary, Alberta, Canada | 10.3 Combustion Air & Fuel Gases |
| 17.1 | Correspondence Course, Lecture 12, Sec. 3, Steam Generation, Water Treatment, S.A.I.T., Calgary, Alberta, Canada | 12.1 Feedwater, Types & Operation |
| 17.2 | Correspondence Course, Lecture 12, Sec. 2, Steam Generation, Water Treatment, S.A.I.T., Calgary, Alberta, Canada | 12.2 Feedwater, Water Treatments |

| Supplementary Packet # | Description | Related Training Module |
|---|---|---|
| 17.3 | Correspondence Course, Lecture 7, Sec. 2, Steam Generators, Boiler Feedwater Treatment, S.A.I.T., Calgary, Alberta, Canada | 12.3 Feedwater, Testing |
| 18.1 | Correspondence Course, Lecture 2, Sec. 5, Electricity, Direct Current Machines, S.A.I.T., Calgary, Alberta, Canada | 11.1 Generators, Types & Construction |
| 18.1<br>18.2 | Correspondence Course, Lecture 4, Sec. 5, Electricity, Alternating Current Generators, S.A.I.T., Calgary, Alberta, Canada | 11.1 Generators, Types & Construction<br>18.2 Generators, Operation |
| 19.1 | Correspondence Course, Lecture 5, Sec. 4, Prime Movers & Auxiliaries, Air Compressor I, S.A.I.T., Calgary, Alberta, Canada | 13.1 Air Compressors, Types |
| 19.1 | Correspondence Course, Lecture 6, Sec. 4, Prime Movers & Auxiliaries, Air Compressors II, S.A.I.T., Calgary, Alberta, Canada | 13.1 Air Compressors, Types<br>13.2 Air Compressors, Operation & Maintenance |
| 20.1 | Basic Electronics, Power Transformers, EL-BE-51 | 15.4 Transformers |
| 21.1 | Correspondence Course, Lecture 6, Sec. 5, Electricity, Switchgear & Circuit, Protective Equipment, S.A.I.T., Calgary, Alberta, Canada | 15.3 Circuit Protection |
| 22.1 | Correspondence Course, Lecture 10, Sec. 3, Prime Movers, Power Plant Erection & Installation, S.A.I.T., Calgary, Alberta, Canada | 15.1 Installation Foundations |

## RECOMMENDATIONS FOR USING TRAINING MODULES

The following pages list modules and their corresponding numbers for this particular apprenticeship trade. As related training classroom hours vary for different reasons throughout the state, we recommend that the individual apprenticeship committees divide the total packets to fit their individual class schedules.

There are over 130 modules available. Apprentices can complete the whole set by the end of their indentured apprenticeships. Some apprentices may already have knowledge and skills that are covered in particular modules. In those cases, perhaps credit could be granted for those subjects, allowing apprentcies to advance to the remaining modules.

We suggest the the apprenticeship instructors assign the modules in numerical order to make this learning tool most effective.

## SUPPLEMENTARY INFORMATION

## ON CASSETTE TAPES

Tape 1: Fire Tube Boilers - Water Tube Boilers and Boiler Manholes and Safety Precautions

Tape 2: Boiler Fittings, Valves, Injectors, Pumps and Steam Traps

Tape 3: Combustion, Boiler Care and Heat Transfer and Feed Water Types

Tape 4: Boiler Safety and Steam Turbines

NOTE: The above cassette tapes are intended as additional reference material for the respective modules, as indicated, and not designated as a required assignment.

Modules 18.1, 19.1, and 20.1 have been omitted because they contain dated materials.

INSTRUCTIONAL LEARNING SYSTEMS

# 8.1

STEAM TURBINES — TYPES

## Goal:

The apprentice will be able to describe the common types of steam turbines.

## Performance Indicators:

1. Describe impulse turbines.
2. Describe reaction turbines.
3. Describe compound turbines.
4. Describe velocity blading.
5. Describe reaction blading.
6. Describe turbine compounding.
7. Describe other types of turbines.

# Study Guide

* Read the goal and performance indicators to find what is to be learned from package.
* Read the vocabulary list to find new words that will be used in package.
* Read the introduction and information sheets.
* Complete the job sheet.
* Complete self-assessment.
* Complete post-assessment.

# Vocabulary

* Compound turbines
* Condensing turbine
* Condensing bleeder turbine
* Cross compound turbine
* Extraction turbines
* Impulse turbines
* Kinetic energy
* Pressure compounding
* Pressure velocity compounding
* Reaction blading
* Reaction turbine
* Tandem compound turbine
* Turbine compounding
* Velocity blading

# Introduction

Turbines are a type of motor that is mounted on a shaft and consists of rotor blades that are actuated by steam, gas, water or other pressure. The steam turbine is common to the American industrial setting.

Basically, turbines operate on the principles of impulse and reaction. Turbine design uses these principles, singly and in combination, to improve the efficiency of turbines.

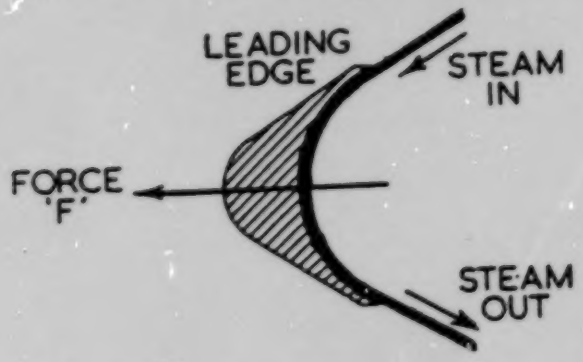
Efficiency involves capturing most of the force or energy of the steam that passes through the turbine blades. Several methods of "Compounding" this energy have been incorporated into the design of turbine engines.

# Information

The steam turbine is used by industries where power and heat is needed to perform processing. Refineries, paper mills, food processing, heating plants and many other industries utilize steam turbines as a prime mover.

Impulse Turbines

The impulse turbine uses stationary steam nozzles to turn a rotor with blades or buckets. Steam is directed at high velocity at the blades. The high velocity of the steam is a result of lowered steam pressure. As steam pressure is reduced the heat energy of the steam is converted into kinetic energy. The blades of the rotor convert the kinetic energy into mechanical energy which turns the rotor and shaft. The impulse principle can be shown.




Section of Turbine Blade

The velocity increases and pressure drops as the steam passes through the nozzles. As the steam passes through the blades, the velocity drops but pressure remains constant. On impulse turbines, pressure at the inlet to blades is the same as that at the outlet from the blades.

Reaction Turbine

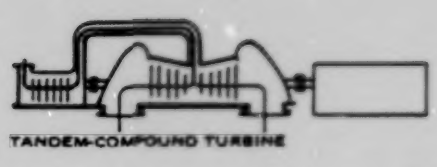
In the pure form, reaction turbines are not used in industry. The reaction turbine that is common uses the principles of impulse and reaction. There are equal numbers of rows of fixed and rotating blades on the rotor. The steam velocity increases as it passes through the fixed blades. As it strikes the fixed blades, the pressure of the steam is reduced and velocity increased due to a change in direction. This produces a driving force in the same manner as on impulse turbine. The steam will next undergo a reaction process. The rotating blades are arranged in a manner that will allow the pressure to drop. Remember, in the impulse turbine the pressure remains the same as it passes through the

# Information

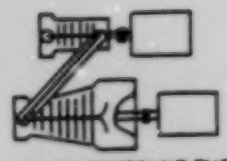
blades. In a reaction turbine, this pressure drop allows more heat energy to be converted into a driving force. The reaction turbine has a driving force equal to the energy converted by impulse and that from the reaction. In both cases, the pressure drop increased velocity; converted heat energy to kinetic energy; and converted kinetic energy into mechanical energy for turning the rotor. Commercial turbines are often a combination of impulse and reaction types.

## Compound Turbines

Compound turbines consist of two or more large turbines linked with one or more generators. A tandem compound turbine is two or more turbines hooked in series with one gerneator. A cross compound turbine is a coupling of turbines in which each is on its own shaft and has its own generator. The steam flows from one to the other.



TANDEM-COMPOUND TURBINE



CROSS-COMPOUND TURBINE

## Velocity Blading

Velocity blading is a method of reducing steam velocity from boiler pressure to exhaust pressure. Steam is expanded within one set of nozzles and routed through rows of moving and fixed blades. The steam velocity is absorbed in the moving blades. The fixed blades act to redirect the steam to the next row of moving blades. Since the velocity is absorbed by several rows of moving blades, the blade speed is less than if the velocity was absorbed by a single row of blades. Velocity blading is a method for reducing the speed of the rotor. For maximum efficiency, the blade velocity should be one half that of the steam velocity.

# Information

Reaction Blading

Reaction blading uses the principles of impulse and reaction turbines to pull the maximum energy from the steam. As steam expands in the fixed blades, velocity increases and the pressure drops. The increased velocity hits the moving blades and exerts a force on the rotor. The steam will continue to increase and decrease pressure and velocity as it passes through alterneting rows of fixed and moving blades. This type of blading "milks" the last drop of energy from the passing steam.

Turbine Compounding

Turbine design attempts to utilize all of the kinetic energy available in steam as it passes through the blades. A number of methods are used to capture the energy. Most are based upon the principle that when steam expands, the velocity increases. Three basic methods for compounding the energy of steam are utilized in the design of turbines.

1. Pressure compounding
2. Velocity compounding
3. Pressure-velocity compounding

Pressure compounding drops the steam pressure in stages. The pressure drops occur as steem passes through nozzles. This system uses several nozzles that are fitted into diaphragms to keep each stage of compounding separate from the others. As the steam moves into each row of blades, the blade speed is reduced. To get the proper blade speed, additional stages can be added.

Velocity compounding uses one set of nozzles and several rows of fixed and moving blades. As steam passes from stationary to moving blades, a change of direction of energy occurs. Velocity is absorbed by more than one row of moving blades. The blade velocity is reduced to a ratio of maximum efficiency. Some small turbines have only single wheels and use return guides or reversing chambers to change direction of steam and force it to give up its useful energy. The singel wheel types are axial re-entry or radial re-entry turbines.

Pressure-velocity Compounding

This system of compounding utilizes the principles of pressure and velocity compounding. Velocity compounded turbines are arranged in series on the same shaft. A number of sets of nozzles are added to give the desired pressure drop.

# Information

## Other Classifications and Types of Turbines

Condensing turbines exhausts steam to a condenser where the heat of the steam is transferred to the cooling water and returned to the boiler. A condensing-bleeder turbine has places for bleeding off the steam at various points. The bleed steam is used for heating the feedwater.

Extraction turbines allow steam to be extracted at one or more points. The extracted steam can be used for purposes other than heating feedwater as in the case of bleeder turbines.

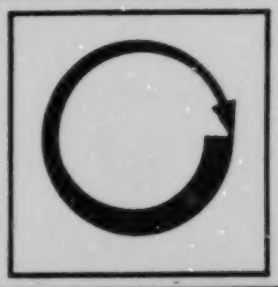
### Condensing Turbines

Condensing turbines operate in conjunction with a condenser. The exhaust pressure is reduced below that of atmospheric pressure. The condenser changes the exhaust steam to water and returns it as boiler feedwater.

### Non-condensing Turbines

Small turbines often discharge the steam exhaust into the atmosphere or uses it as process steam. When steam is not returned to the boiler it is called a non-condensing turbine. Condensers are not part of a non-condensing turbine.

# Assignment



* Read pages 1 - 12 in supplementary reference.
* Complete job sheet.
* Complete self-assessment and check answers with answer sheet.
* Complete post-assessment and ask instructor to check your answers.

# Job Sheet

ANALYZE TURBINE SPECIFICATIONS

* Obtain turbine specifications from a supply catalog, equipment manual or other source.
* Read and analyze the specifications of a specific model turbine.
  - Is the turbine an impulse or reaction turbine?
  - Is it a condensing, non-condensing, extraction, bleeder turbine?
  - How is the turbine compounded?
* Ask instructor to explain those features that you do not understand.

# •Self Assessment

Match the following turbine terms with appropriate phrases.

| | |
|---|---|
| 1. Impulse turbine | A. Method of reducing steam velocity. |
| 2. Tandem compound turbine | B. Drops steam pressure in stages. |
| 3. Reaction turbine | C. Exhausts steam to a condenser. |
| 4. Cross compound turbine | D. Converted from heat energy as pressure of steam is reduced. |
| 5. Velocity compounding | E. Energy that turns the rotor. |
| 6. Pressure compounding | F. Stationary steam nozzles. |
| 7. Mechanical energy | G. Allows steam to be extracted at various points in turbine. |
| 8. Condensing turbine | H. Not used in pure form by industry. |
| 9. Kinetic energy | I. Two or more turbines in series with one generator. |
| 10. Extraction turbine | J. Two or more generators on their own shaft and with their own generator but using the same steam. |

# Self Assessment Answers

1. F
2. I
3. H
4. J
5. A
6. B
7. E
8. C
9. D
10. G

# Post Assessment

Mark the following statements (T or F) true or false.

1. Reaction turbines in their pure form are not used in industry. _____

2. The so called reaction turbines use both impulse and reaction principles. _____

3. As steam expands in the nozzles pressure drops. _____

4. Velocity increases as steam pressure is reduced. _____

5. A cross-compound turbine is a series of turbines mounted on a single shaft with one generator. _____

6. Velocity blading is a method of increasing the speed of the rotor. _____

7. Velocity compounding reduces steam pressure through a series of stages or nozzles that are separated by diaphragms. _____

8. Condensing turbines return their exhaust steam as feedwater to the boiler by running it through a condenser. _____

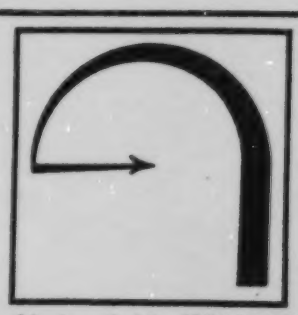
9. Extraction turbines allow steam to be pulled from the turbine for other purposes such as steam cleaning. _____

10. Axial re-entry turbines are a large type of turbine. _____

# Instructor Post Assessment Answers

1. T
2. T
3. T
4. T
5. F
6. F
7. F
8. T
9. T
10. F

# Supplementary References



* Correspondence Course. Lecture 1, Section 4, Third Class. Southern Alberta Institute of Technology. Calgary, Alberta, Canada.

SOUTHERN ALBERTA INSTITUTE OF TECHNOLOGY
CALGARY, ALBERTA

Correspondence Courses
Power Engineering

SECTION 4

Third Class
Lecture 1

PRIME MOVERS AND AUXILIARIES

# STEAM TURBINES

## INTRODUCTION

Of all heat engines and prime movers the steam turbine is nearest to the ideal and it is widely used in power plants and in all industries where power and/or heat is needed for processes. These include: pulp mills, refineries, petro-chemical plants, food processing plants, desalination plants, refuse incinerating and district heating plants.

### Advantages

The successful application in so many industries is due to the many ideal features of the steam turbine. These features include:

1. Ability to utilize high pressure and high temperature steam.
2. High efficiency.
3. High rotational speed.
4. High capacity/weight ratio.
5. Smooth, nearly vibration-free operation.
6. No internal lubrication.
7. Oilfree exhaust steam.
8. Can be built in small or very large units ( up to 1200 MW ).

## Disadvantages

The few disadvantages of the steam turbine are:

1. For slow speed application reduction gears are required.
2. The steam turbine cannot be made reversible.
3. The efficiency of small simple steam turbines is poor.

## OPERATION PRINCIPLES

### Impulse Turbine

In principle the impulse steam turbine consists of a casing containing stationary steam nozzles and a rotor with moving or rotating buckets.

The steam passes through the stationary nozzles and is directed at high velocity against the rotor buckets causing the rotor to rotate at high speed.

The following events take place in the nozzles:

The steam pressure decreases.

The enthalpy of the steam decreases.

The steam velocity increases.

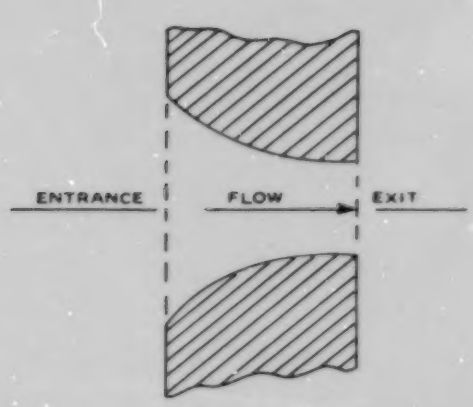
The volume of the steam increases.

There is a conversion of heat energy to kinetic energy as the heat energy from the decrease in steam enthalpy is converted into kinetic energy by the increased steam velocity.

The nozzles may be convergent nozzles ( Fig. 1 ) or they may be convergent-divergent nozzles ( Fig. 2 ). Convergent nozzles are used for smaller pressure drops where the minimum exit pressure is 0.577 x the inlet pressure ( the critical pressure for nozzles. ).

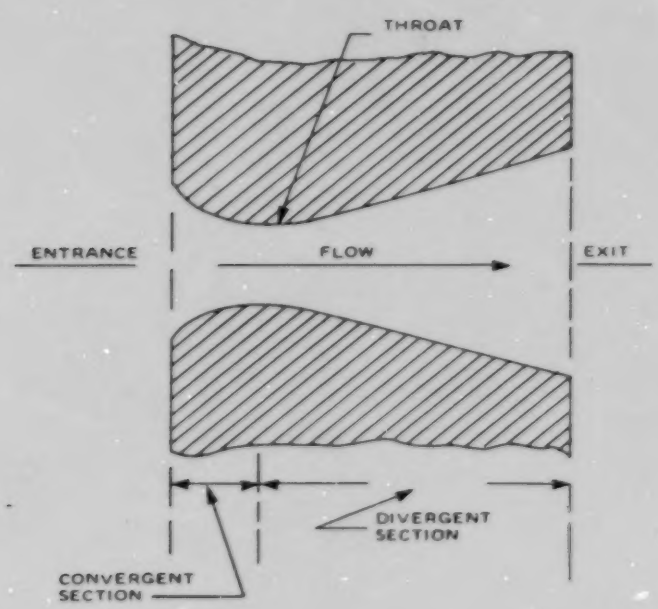
If the exit pressure is less than 0.577 x inlet pressure, eddy-currents are developed and the exit velocity will be less than calculated.

The convergent-divergent nozzles prevent eddy-currents and the calculated velocity will be obtained even at large pressure drops.




Convergent Nozzle

Fig. 1




Convergent-Divergent Nozzle

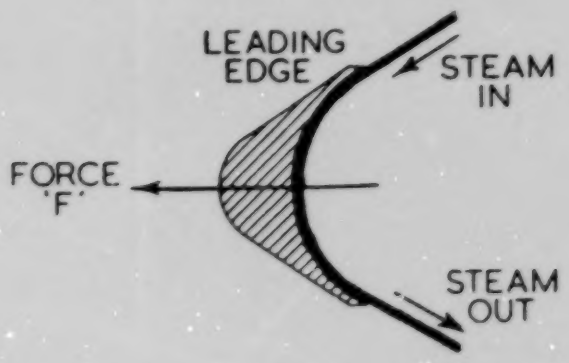
Fig. 2

The purpose of the bucket or moving blade on the rotor is to convert the kinetic energy of the steam into mechanical energy. If all kinetic energy is converted the steam exit velocity will be 0 m/s. This is not possible but it shows that the rotor blades must bring the steam exit velocity near 0 m/s.

( PE3-4-1-3)

The Impulse Principle

If steam at high pressure is allowed to expand through a stationary nozzle, the result will be a drop in the steam pressure and an increase in steam velocity. In fact, the steam will issue from the nozzle in the form of a high speed jet. If this high velocity steam is applied to a properly shaped turbine blade, it will change in direction due to the shape of the blade ( Fig. 3 ). The effect of this change in direction of the steam flow will be to produce an impulse force, marked F in Fig. 3, on the blade causing it to move. If the blade is attached to the rotor of a turbine, then the rotor will revolve.




Section of Turbine Blade

Fig. 3

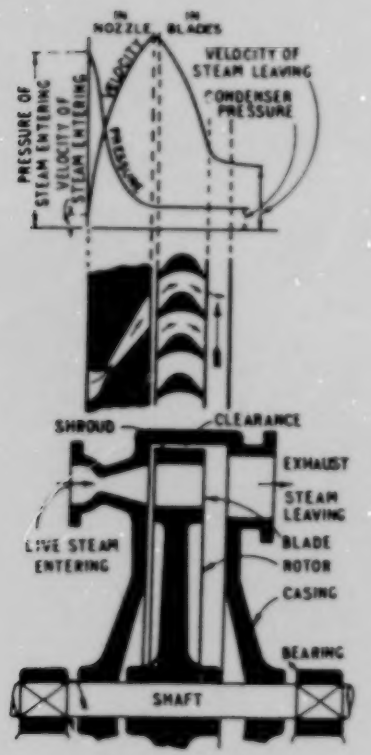
In Fig. 3, force applied to the blade is developed by causing the steam to change direction of flow ( Newton's 2nd Law - change of momentum ). The change of momentum produces the impulse force.

In an actual impulse turbine there are a number of stationary nozzles and the moving blades are arranged completely around the rotor periphery.

Fig. 4 shows the nozzle and blade arrangement in a simple impulse turbine and the graph in the figure indicates how the pressure and the velocity of the steam change as the steam passes through first the stationary nozzles and then the moving blades.

Note that the pressure drops and the velocity increases as the steam passes through the nozzles. Then as the steam passes through the moving blades the velocity drops but the pressure remains the same.

The fact that the pressure does not drop across the moving blades is the distinguishing feature of the impulse turbine. The pressure at the inlet to the moving blades is the same as the pressure at the outlet from the moving blades.



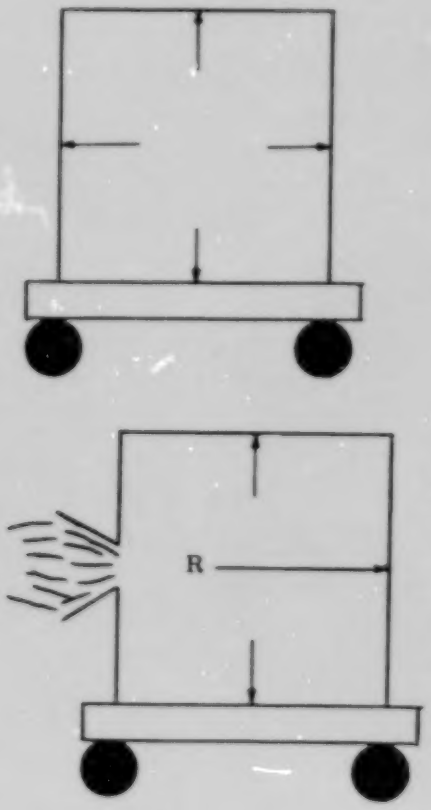

Simple Impulse Turbine

Fig. 4

If there is no escape opening or nozzle for the steam, then the pressure will be the same on all walls of the container and the container will remain at rest. If, however, the container has an escape opening or nozzle, then steam will expand through the opening and drop in pressure. As a result there will be an unbalanced pressure on the wall opposite to the opening and a reaction force R will be produced causing the container to move.

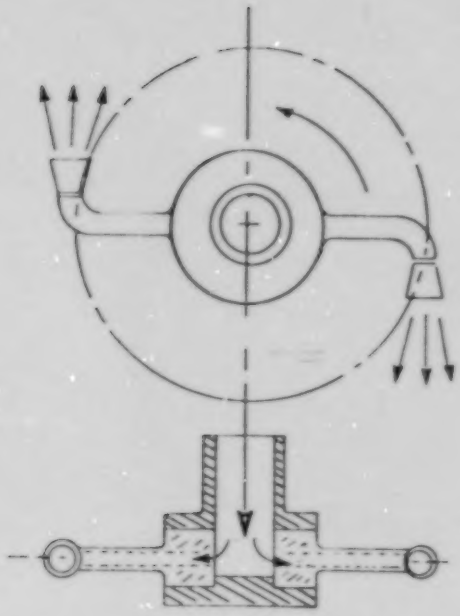
## The Reaction Principle

If the moving blades of a turbine are shaped in such a way that the steam expands and drops in pressure as it passes through them, then a reaction will be produced which gives a force to the blades. This reaction effect can be illustrated by considering a container filled with high pressure steam as in Fig. 5.




Reaction Effect

Fig. 5



Pure Reaction Turbine

Fig. 6

Fig. 6 shows a diagram of this principle applied to a turbine drive.

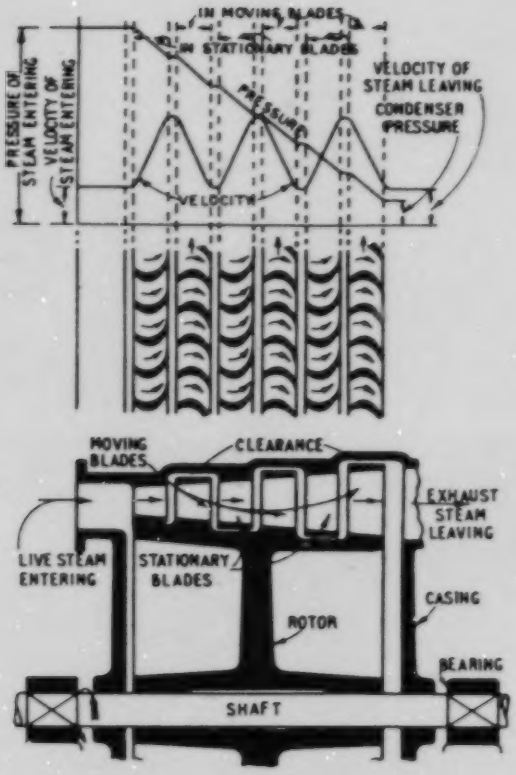
A reaction turbine has rows of fixed blades alternating with rows of moving blades. The steam expands first in the stationary or fixed blades where it gains some velocity as it drops in pressure. It then enters the moving blades where its direction of flow is changed thus producing an impulse force on the moving blades. In addition, however, the steam upon passing through the moving blades, again expands and further drops in pressure giving a reaction force to the blades.

This sequence is repeated as the steam passes through additional rows of fixed and moving blades.

Fig. 7 shows the blade arrangement and the pressure and velocity changes of the steam in a reaction turbine.

Note that the steam pressure drops across both the fixed and the moving blades while the absolute velocity rises in the fixed blades and drops in the moving blades.

The distinguishing feature of the reaction turbine is the fact that the pressure does drop across the moving blades. In other words there is a pressure




Simple Reaction Turbine

Fig. 7

difference between the inlet to the moving blades and the outlet from the moving blades.

Special Aspects of Reaction Turbines

1. There is a difference in pressure across the moving blades. The steam will therefore tend to leak around the periphery of the blades instead of passing through them. Blade clearances therefore must be kept to a minimum.

2. Also, due to pressure drop across the moving blades, an unbalanced thrust will be developed upon the rotor and some arrangement must be made to balance this.

## Impulse Turbine Staging

In order for the steam to give up all its kinetic energy to the moving blades in an impulse turbine, it should leave the blades at zero velocity. This condition will exist if the blade velocity is equal to one half of the steam velocity. Therefore, for good efficiency the blade velocity should be about one half of the steam velocity.

If the steam was expanded from admission pressure down to final exhaust pressure in a single set of nozzles ( single stage ) then the velocity of the steam leaving the nozzles might be in the order of 1100 m per s. In order to have good efficiency the blade velocity would have to be about 550 m per s., which would require excessively high rev/min of the turbine rotor and failure due to centrifugal force could result.

In addition to this objection, excessively high steam velocity will cause high friction losses in nozzles and blading.

In order to reduce steam velocity and blade velocity, the following methods may be used:

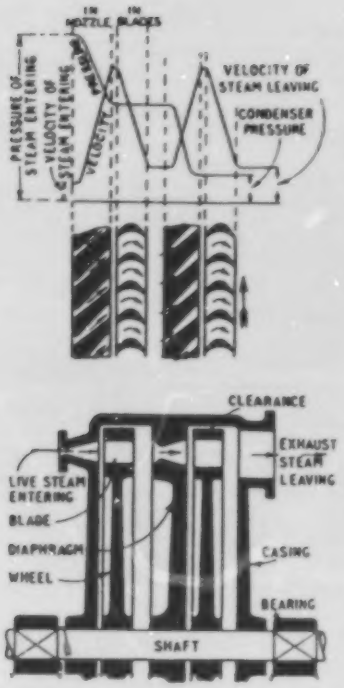
1. Pressure compounding,
2. Velocity compounding,
3. Pressure-velocity compounding.

### 1. Pressure Compounding

The expansion of steam from boiler pressure to exhaust pressure is carried out in a number of steps or stages. Each stage has a set of nozzles and a row of moving blades. The rows of moving blades are separated from each other by partitions or diaphragms into which the nozzles are set. As only a portion of the velocity available is developed in each set of nozzles, the blade velocity is kept down to a reasonable amount.

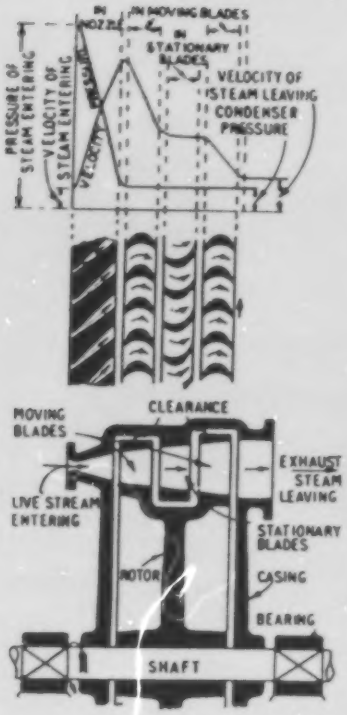
This type of compounding is known as the Rateau method and the nozzle and blade arrangement for a pressure compounded impulse turbine is sketched in Fig. 8.

In this arrangement, the pressure of the steam drops in each set of nozzles as indicated by the pressure graph. The steam velocity is increased by each pressure drop and then decreases again in each row of moving blades, as the velocity graph shows.




Pressure Compounded Impulse Turbine

Fig. 8




Velocity Compounded Impulse Turbine

Fig. 9

2. Velocity Compounding

This design consists of one set of nozzles in which the steam is expanded from initial to exhaust pressure. The velocity of the steam resulting from this expansion is absorbed in two or more rows of moving blades. Rows of fixed or guide blades, attached to the casing, are set between the rows of moving blades and receive and redirect the steam to the next row of moving blades. As the velocity is absorbed in more than one row of moving blades, the blade speed is less than if the velocity was all absorbed in one row of blades.

This type of compounding is known as the Curtis method and the blade and nozzle arrangement for a velocity compounded impulse turbine is shown in Fig. 9.

( PE3-4-1-9 )

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The pressure drops from inlet pressure to exhaust pressure in the single set of nozzles as the pressure graph shows. This large single pressure drop produces high steam velocity which is absorbed in the two rows of moving blades. Note that there is no pressure or velocity drop in the fixed guide blades.

3. Pressure-Velocity Compounding

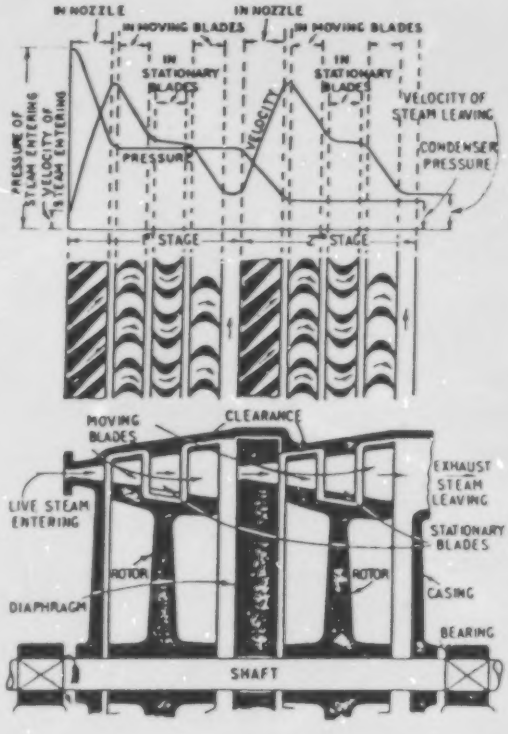
This is a combination of the first two methods of compounding, namely pressure compounding and velocity compounding.

The steam is expanded in two or more sets of nozzles in series, each set having velocity compounded blades to receive the steam issuing from the nozzles.

The arrangement shown in Fig. 9 ( a ) features two sets of nozzles. The steam pressure drops in each set of nozzles and the resulting velocity increase in each case is absorbed by two rows of moving blades having a row of stationary blades between them.

The methods of reducing rotor speeds, namely, pressure compounding, velocity compounding, and pressure-velocity compounding have all applied to impulse turbines.

In the case of the reaction turbine, it is not necessary to make special blade arrangements to reduce rotor speed. This is because the pressure drops across each row of moving blades as well as across each row of fixed blades and consequently the pressure drops in even stages and small amounts all through the machine. This requires, however, a large number of alternate rows of fixed and moving blades resulting in a long machine. Therefore, in order to reduce the number of blade rows necessary, reaction turbines frequently have a velocity compounded impulse stage at the inlet end of the machine.




Pressure-Velocity Compounded Impulse Turbine

Fig. 9 ( a )

( PE3-4-1-10 )

## TYPES OF TURBINES

### Condensing Turbines

With the condensing turbine the steam exhausts to the condenser and the latent heat of the steam is transferred to the cooling water. The condensed steam is returned to the boiler as feedwater.

### Condensing-Bleeder Turbines

The condensing-bleeder turbine reduces the condenser losses as steam is bled off at several points of the turbine. The bleed-steam is used for feedwater heating; up to 20% of the total steam flow may be bled off.

### Back-Pressure Turbines

Back-pressure turbines are often used in industrial plants, the turbine acts as a reducing station between boiler and process steam header. The process steam pressure is kept constant and the generator output depends on the demand for process steam.

The back-pressure turbine may also have bleed points and is then called a back-pressure-bleeder-turbine.

### Extraction Turbines

Extraction turbines are turbines where steam is extracted at one or more points at constant pressure.

Extraction turbines may be single or double-extraction-condensing turbines or single-or double-extraction back-pressure turbines. The extraction turbines may, besides extraction points, have bleed points for feedwater heating.

### Topping Turbines

Topping turbines have been used when old boilers are replaced with new high pressure boilers. The turbine is a back-pressure turbine exhausting to the old boiler header still supplying steam to the old lower pressure turbines.

### Mixed Pressure Turbines

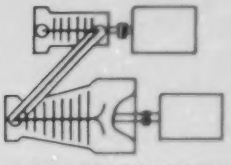
Mixed pressure turbines are used where excess steam from process is available for the low pressure part of the turbine, while steam at boiler pressure may be added to the high pressure part of the turbine when more load is applied to the turbine.

Cross Compound Turbines

Cross compound turbines are large turbines with parallel shafts with a generator on each shaft. The steam flows through the high pressure turbine, then is crossed-over to the low pressure turbine. ( Figs. 10 and 11 )

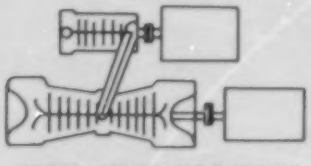
Tandem Compound Turbines

Tandem compound turbines are large turbines consisting of two or more turbines in series coupled together as one shaft and applied to one generator. ( Fig. 12 )



CROSS-COMPOUND TURBINE

Fig. 10



CROSS-COMPOUND TURBINE WITH DOUBLE-FLOW LOW-PRESSURE TURBINE

Fig. 11

TANDEM-COMPOUND TURBINE

Fig. 12

TURBINE PARTS

Turbine Casings

Casings or cylinders are of the horizontal split type. This is not ideal as the heavy flanges of the joints are slow to follow the temperature changes of the cylinder walls. But for assembling and inspection purposes there is no other solution.

The casings are heavy in order to withstand the high pressures and temperatures. It is general practice to let the thickness of walls and flanges decrease from inlet- to exhaust-end.

Large casings for low-pressure turbines are of welded plate construction, while smaller L.P. casings are of cast iron, which may be used for temperatures up to 230° C. Casings for intermediate pressures are generally of cast carbon steel able to withstand up to 425° C. The high temperature high-pressure casings for temperatures exceeding 550° C are of cast alloy steel such as 3 Cr 1Mo (3% Chromium + 1% Molybdenum.)

The reason for using different casing materials is that materials at the given maximum temperatures and under constant pressure continue to deform with very slowly increasing strain of the material; this phenomenon is called "Creep".

The casing joints are made steam tight, without the use of gaskets, by matching the flange faces very exactly and very smoothly. The bolt holes in the flanges are drilled for smoothly fitting bolts, but dowel pins are often added to secure exact alignment of the flange joint. The assembled casing is then machined off inside on a boring-mill, where grooves are made for the diaphragms ( for impulse turbines ) or for the stationary blades ( reaction turbines ). Borings are also made for shaft seals and in many cases for the bearings also.

For high pressures the flanges of the casings must be very heavy and will heat up much slower than the casing walls. Flange heating, by steam through machined channels between the flanges or holes drilled axially through the upper and lower flanges, is often applied. ( Fig. 13 ).

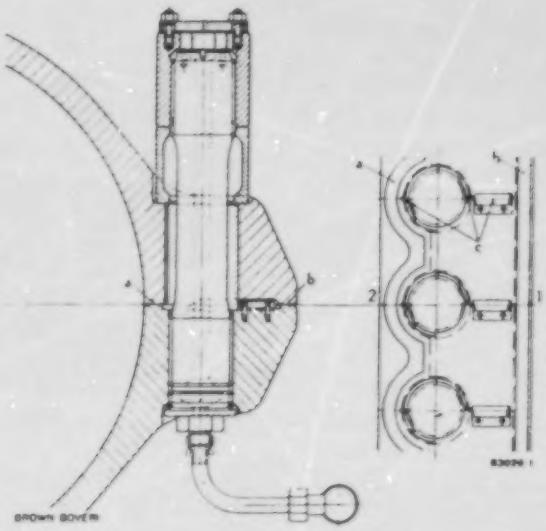



Fig. 13. — Flange warming

The hot steam flows through the space between the bearing surfaces of the flange and is forced to flow round the bolt by means of baffle plates

a, b = Flange bearing surfaces
c = Baffle plates
1 = Outside
2 = Inside

Fig. 13

Double casings are used for very high steam pressures. ( Figs. 15 and 16 ) The high pressure is applied to the inner casing, which is open at the exhaust end, letting the turbine exhaust to the outer casings; the pressure is divided between the casings, and most important, the temperature is also divided and thermal stresses on casings and flanges are greatly reduced. Radiation losses are also decreased. The inner casing may be assembled with shrink rings giving an ideal casing without flanges ( Figs. 17 and 18 ).



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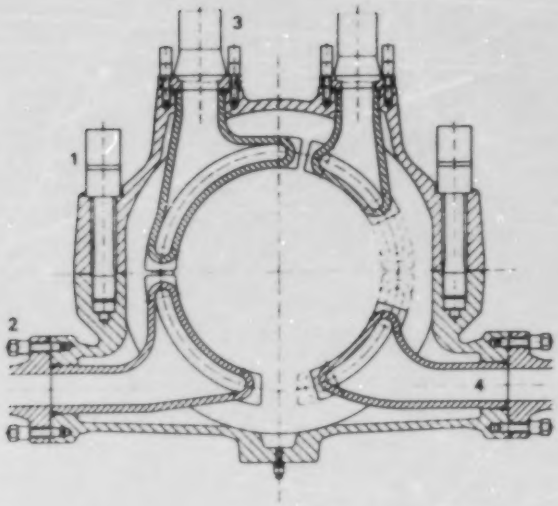
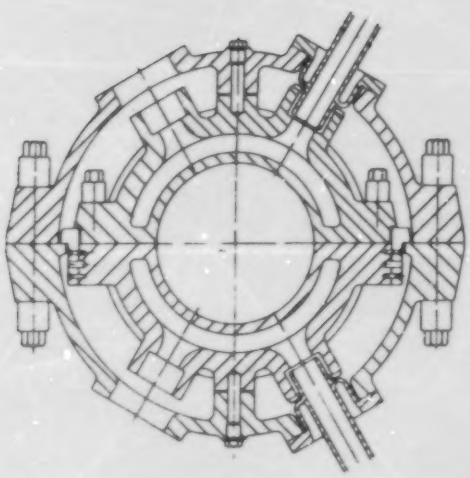



Cross-section through a valve casing with four valves and the wheel-chamber of an industrial turbine.

Fig. 14



Cross-section through Double-shell Hp Turbine

Fig. 15

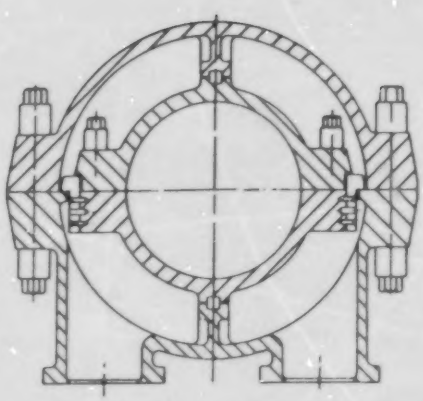


Fig. 16

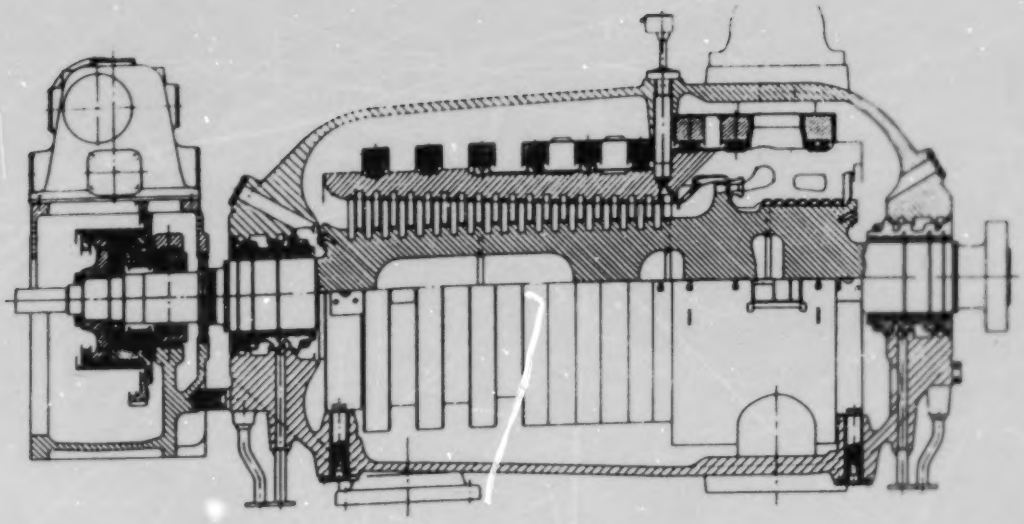
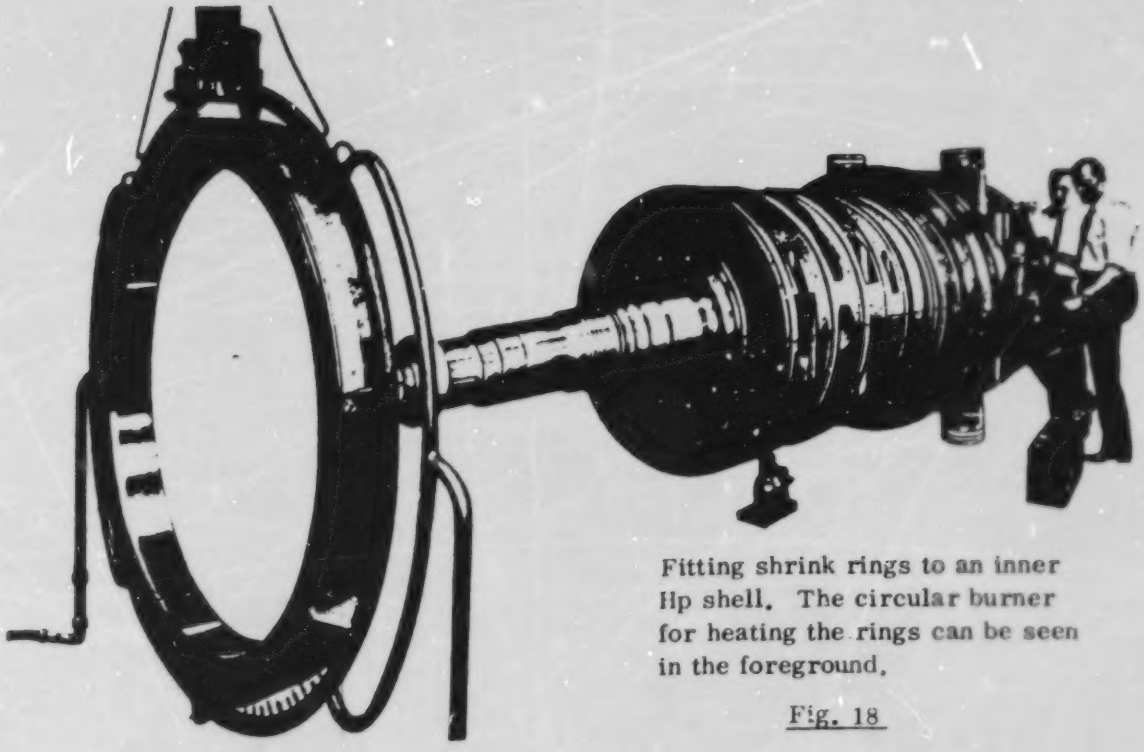


Fig. 17



Fitting shrink rings to an inner Hp shell. The circular burner for heating the rings can be seen in the foreground.

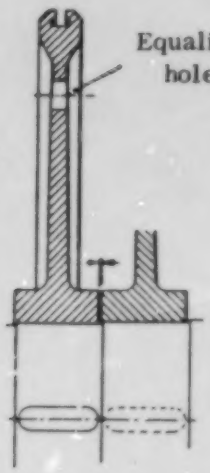
Fig. 18

( PE3-4-1-15 )

Turbine Rotors

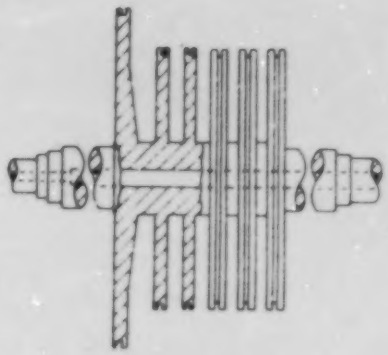
The design of a turbine rotor depends on the operating principle of the turbine.

The impulse turbine with pressure drop across the stationary blades must have seals between stationary blades and the rotor. The smaller the sealing area, the smaller the leakage; therefore the stationary blades are mounted in diaphragms with labyrinth seals around the shaft. This construction requires a disc rotor ( Figs. 19 and 20 ).




Rotor with shrunk-on discs

Fig. 19



Solid Forged Disc Rotor

Fig. 20

The reaction turbine has pressure drops across the moving as well as across the stationary blades and the use of a disc rotor would create a large axial thrust across each disc. The application of a drum rotor eliminates the axial thrust caused by the discs, but not the axial thrust caused by the differential pressure across the moving blades.

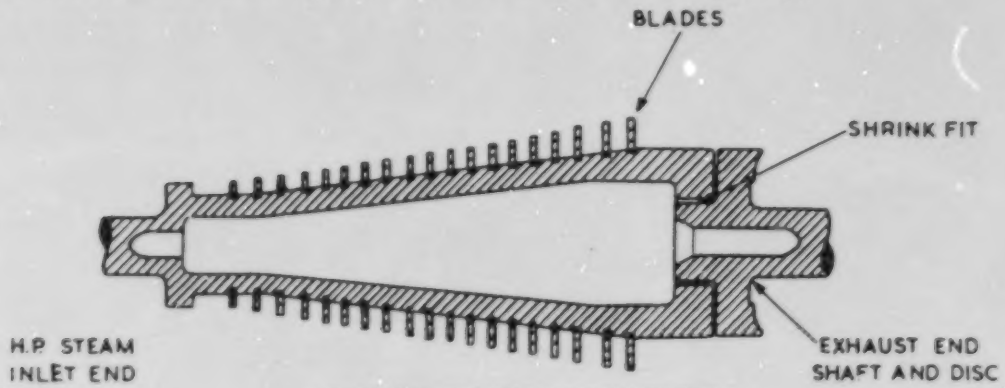
1. Disc Rotors

All larger disc rotors are now machined out of a solid forging of nickel-steel; this should give the strongest rotor and a fully balanced rotor. It is rather expensive, as the weight of the final rotor is approximately 50% of the initial forging. ( Fig. 20 ) Older or smaller disc rotors have shaft and discs made in

separate pieces with the discs shrunk on the shaft. The bore of the discs is made 0,1% smaller in diameter than the shaft. The discs are then heated until they easily are slid along the shaft and located in the correct position on the shaft and shaft key. A small clearance between the discs prevents thermal stress in the shaft.

2. Drum Rotors

The first reaction turbines had solid forged drum rotors. They were strong, generally well balanced as they were machined over the total surface. With the increasing size of turbines the solid rotors got too heavy and the hollow drum rotor was introduced. This rotor is made of two or more pieces. For good balance the drum must be machined both outside and inside and the drum must be open at one end. The second part of the rotor is the drum end cover with shaft. The end cover is made with a shrink fit and welded ( Fig. 21 ).




Hollow Drum Rotor

Fig. 21

A fairly light and rigid drum rotor may be manufactured from discs welded together to form a drum as shown in Fig. 22. Before welding, the rotor is heated by induction heating, then the welding is performed with automatic welding machines for the "Argon-arc "process, where the arc burns in an argon atmosphere.

Most rotors are now made of nickel alloy-steels with elastic limits of around $300 \times 10^6$ pascals. Rotors for high outputs and high temperatures are generally made of chromium-nickel-molybdenum steels with elastic limits of $(600 - 700) \times 10^6$ pascals.

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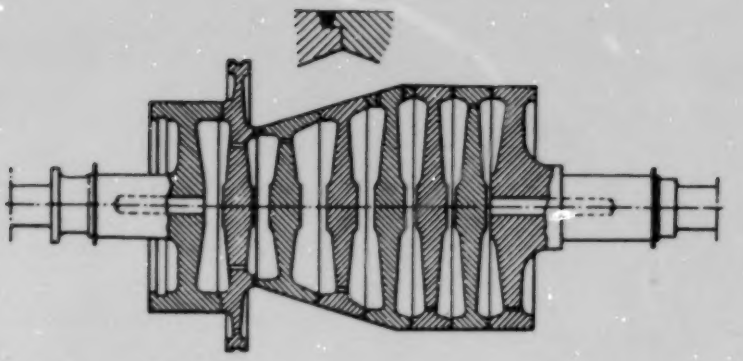


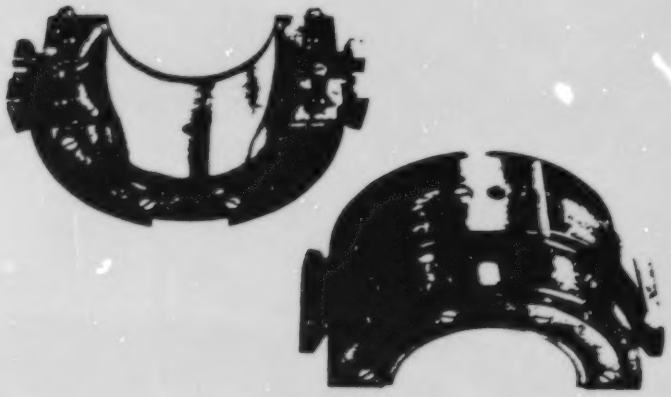
Brown-Boveri Welded Turbine Rotor

Fig. 22

Turbine Bearings

1. Journal Bearings

The bearings for small turbines are often self-aligning spherical ball or roller-bearings or they may be ring lubricated sleeve bearings with bronze or babbitt lining ( Figs. 23 and 24 ).



Babbitt-lined Journal Bearing

Fig. 23

Ring-oiled Sleeve Bearing with Radial Babbitt Facing

Fig. 24

( PE3-4-1-18 )

2. Thrust Bearings

The main purposes of the thrust bearing are:

1. to keep the rotor in an exact position in the casing.
2. to absorb any axial thrust on the rotor.

From the thrust bearing the shaft must be free to expand in either direction, thus a shaft can have only one thrust bearing. The thrust bearing should be located at the steam inlet, where the blade clearances are most critical. When shafts of a tandem compound turbine are joined together with solid couplings, only one thrust bearing can be applied. If flexible couplings take up the axial expansion, each shaft must have a thrust bearing ( Fig. 25 ).

Journal Bearing with Kingsbury Thrust Bearing

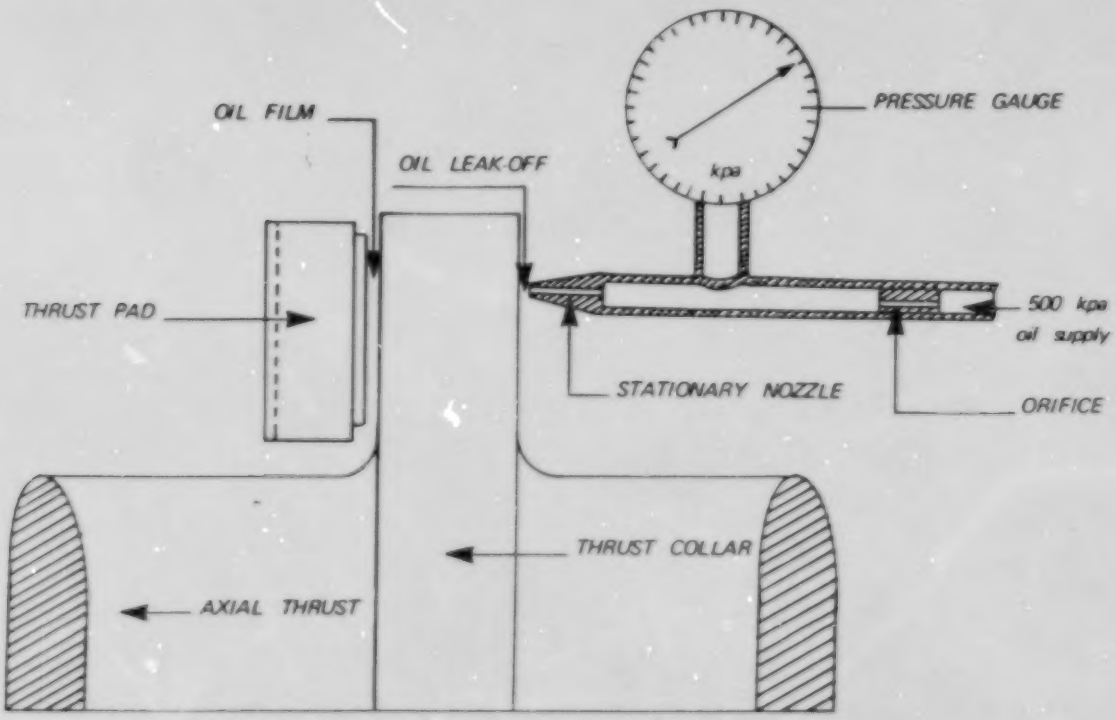
Fig. 25

The axial thrust is very small for impulse turbines as the pressure is equal across the rotor discs ensured by equalizing holes in the discs. A simple thrust bearing such as a ball bearing for small turbines and radial babbitt facing on journal bearings for larger turbines is very common ( Fig. 24 ).

The pressure drop across the moving blades of reaction turbines creates a heavy axial thrust in the direction of steam flow through the turbine and a thrust bearing suitable for heavy axial loading is needed. The tilting pad Kingsbury or Michel thrust bearings operating on the same principle as the tilting pad journal bearing are generally applied. The axial thrust in impulse turbines does not require tilting pad thrust bearings, but due to their excellent performance they

are the most common thrust bearing for large impulse turbines. The axial thrust in reaction turbines can be nearly eliminated by the use of balance or dummy pistons. With the correct size of a dummy piston exposed to two different bleed point pressures, the thrust is nearly equalized. There is a small leakage across the labyrinth seal of the dummy piston as steam leaks from the high to the lower bleed point ( Fig. 27 ).

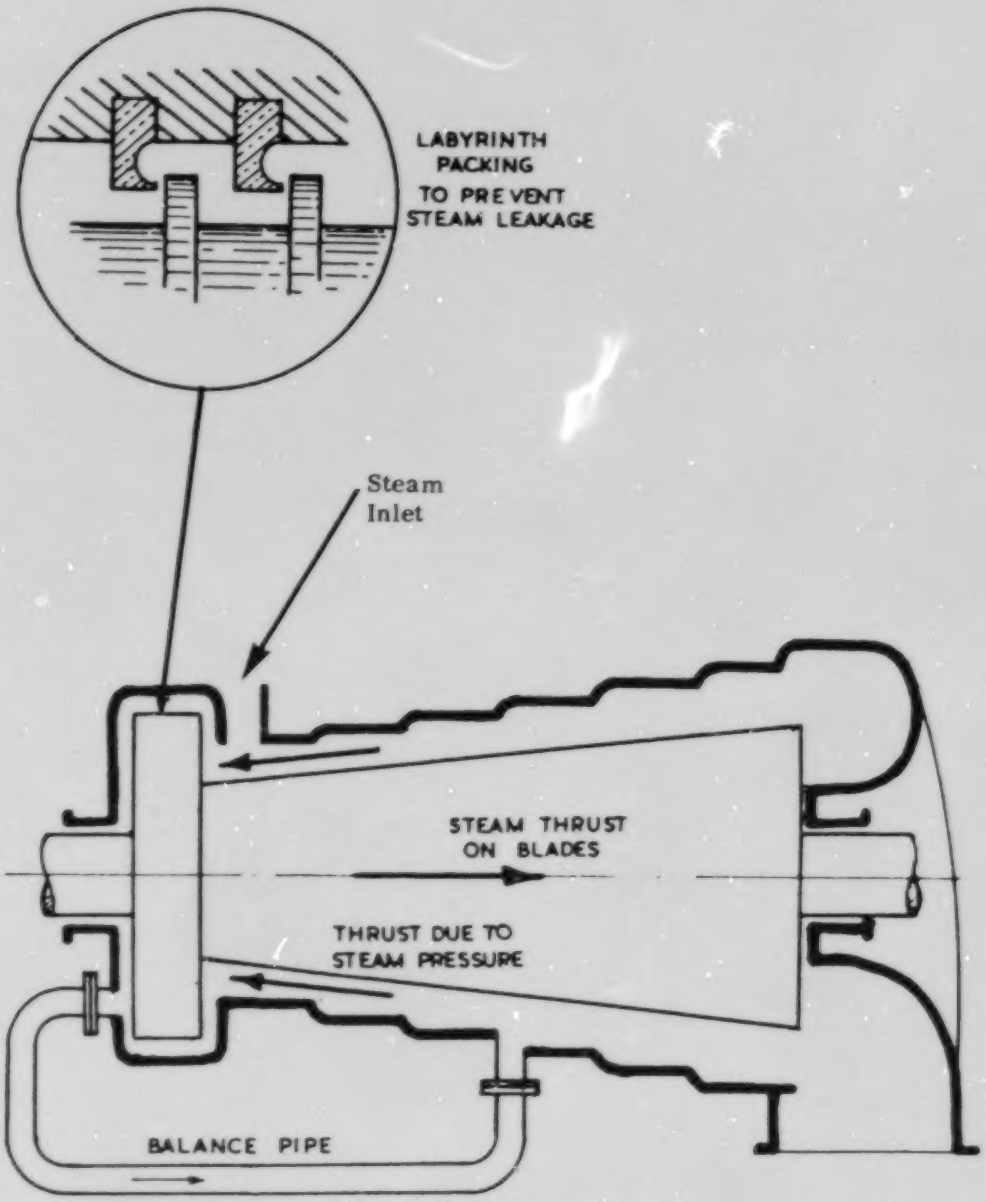
The axial position of the rotor is very important and an axial position indicator is often applied to the thrust bearing. It may be a large dial micrometer with alarm setting for an axial movement of 0.4 millimetre and shutdown at 0.8 millimetre. An oil pressure gage connected to an oil leak-off device may also be used as an axial position indicator ( Fig. 26 ).




Thrust Collar Position Indicator

Fig. 26

( PE3-4-1-20 )




Dummy Piston and Balance Pipe

Fig. 27

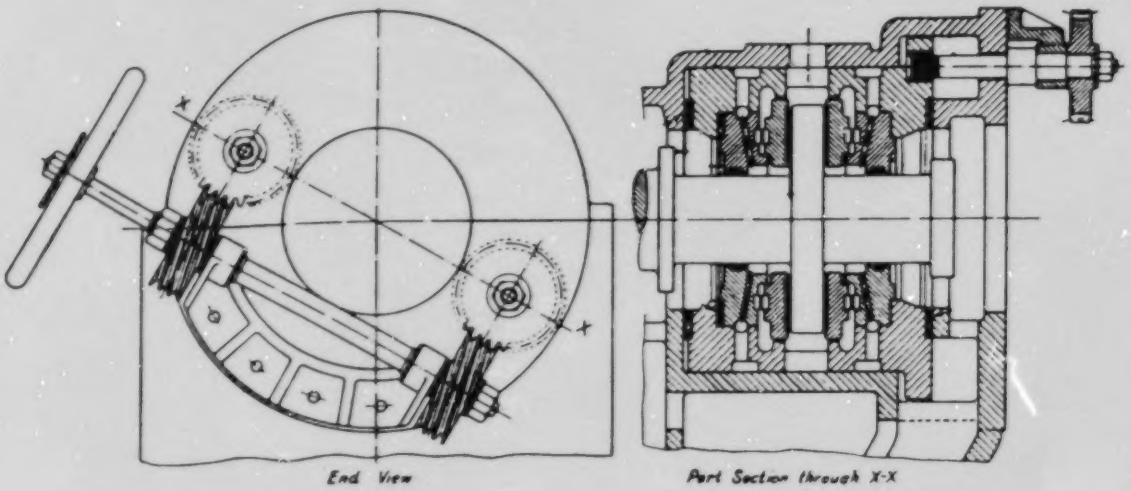
( PE3-4-1-21 )

With the position indicator shown in Fig. 26, the oil is supplied at say 500 kPa, flows through an orifice and leaks off through a nozzle. The pressure between orifice and nozzle depends on the distance between the nozzle and shaft thrust collar; the larger the distance the lower the pressure. The pressure gage can be calibrated in millimetre clearance and may have alarm and shutdown settings.

## Turbine Seals

### 1. Blade Seals

The efficiency of reaction turbines depends to a large extent on the blade seals; radial as well as axial seals are often part of the shroud with the seal clearances kept as small as possible. As protection for the axial seals some manufacturers apply an adjustable thrust bearing ( Fig. 28 ). The whole thrust block is cylindrical and fits like a piston in the cylinder with the whole thrust block able to be axially adjusted as shown in Fig. 28. During startup the thrust block is pushed against a stop in the direction of exhaust for maximum seal clearances. When the turbine is heated up and has been on load for a short time the thrust block is pulled forward against a forward stop for minimum seal clearance and maximum blade efficiency.



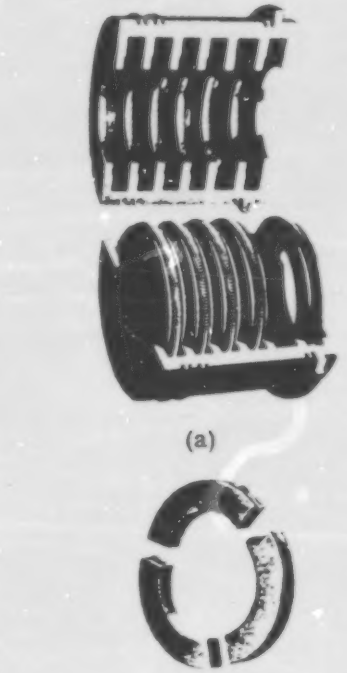

Thrust Adjusting Gear for Reaction Turbine

Fig. 28

## 2. Shaft Seals

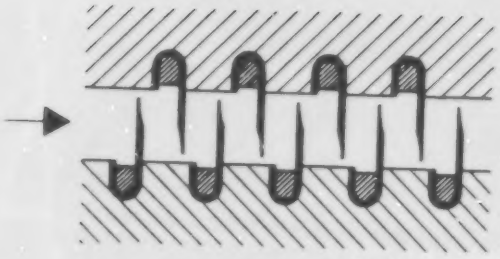
Shaft seals must be provided in order to prevent or at least reduce steam leakage where the shafts extend through the casings. Also when low pressure turbines are under vacuum the seals must prevent air from leaking into the casing.

Ordinary soft packing may be used for shaft sealing in small turbines. Carbon rings ( Fig. 29 are also very common for small turbines. The carbon ring is made up of three segments butting together tightly under the pressure of a garter spring. The ring has a few hundreds of a millimetre clearance around the shaft and is prevented from turning by a locking pin. The ring has a slight side clearance in the housing allowing it to move freely in radial directions. Carbon rings are self-lubricating but have a tendency to corrode the shaft when the turbine is shut down. The presence of moisture accelerates the corrosion. The carbon rings are from 10 to 25 millimetres wide and may cause heating when they ride on the shaft. They are, for that reason, limited to shafts less than 150 millimetres in diameter. For larger diameter shafts where the surface speed is high, labyrinth seals are applied. The labyrinth seal consists of a number of rings 1 - 2 millimetres thick fixed to the shaft, tapered at the outer periphery to nearly knife-sharp with a clearance to the casing of a few hundreds of a millimetre. The rings are of brass or stainless steel, the sharp edge gives better sealing and rubs off easily without excessive heating in case of a slightly eccentric shaft. Some labyrinth seals are very simple, others are complicated. ( Figs. 30, 31 and 33 )



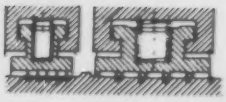

Carbon Ring Seal

Fig. 29



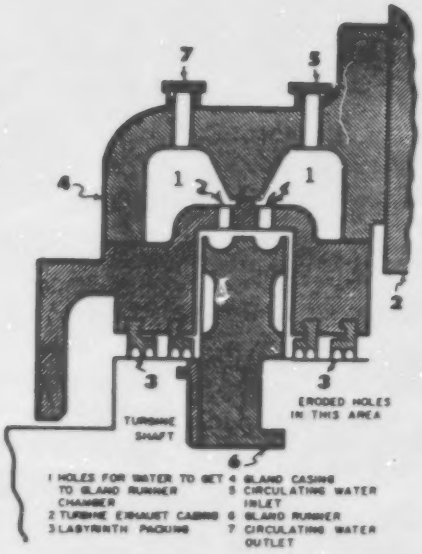
Labyrinth Seal

Fig. 30



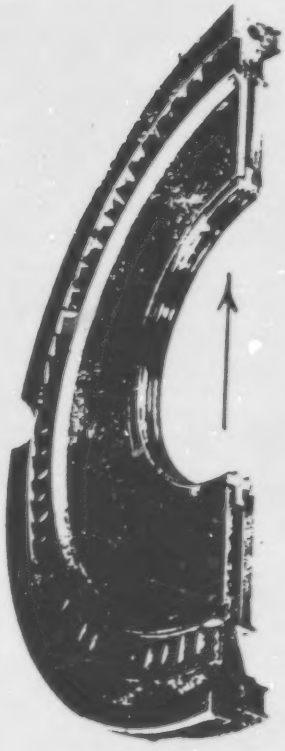
Labyrinth Seal

Fig. 31




Water Sealed Gland

Fig. 32



Diaphragm with Spring Loaded Laybrinth Seal

Fig. 33



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High pressure turbines operating at 12 000 to 15 000 kPa cause a sealing problem, as a straight labyrinth seal for that pressure would be extremely long or have lots of steam leaking through. The problem is solved by a series of steam pockets between sets of labyrinth seals. The high pressure steam leaks through 100 - 200 millimetres of labyrinth seal into the first pocket, which is connected to the H.P. exhaust, thus any steam leaking through the seal is used in the I.P. Turbine. After the first pocket the steam leaks through the second seal 75 - 150 millimetres long and into the second pocket connected to an H.P. feedwater heater. Then steam leaks through the third labyrinth seal to the third pocket connected to the I.P. exhaust. The steam then leaks through the fourth seal into the fourth pocket, which is connected to the L.P. shaft seals supplying them with sealing steam. On the steam line between pocket number 4 and the L.P. seals are two connections with pressure control valves. One is a spill-over valve to the condenser, which will open to the condenser if the gland steam exceeds the set point of a few centimetres of water above atmospheric pressure. The other connection has a control valve to supply gland steam when the pressure decreases to near atmospheric pressure. This valve operates during start-ups and low loads.

Neither the carbon nor the labyrinth shaft seals prevent all leakage. If a positive or leak-proof seal is needed, a water seal may be installed. The water seal consists of an impeller on the turbine shaft which rotates in a waterfilled casing and water thrown out from the impeller forms a leak-proof water barrier ( Fig. 32 ). Water seals are mainly applied to L.P. glands to guard against air leakage, but they may also be applied as the final seal for H.P. and I.P. glands.

The water seal cannot operate properly at low speed and gland steam must be applied for sealing during start-up until the turbine speed is approximately 2000 rev/min. Water seals are supplied with clean cool condensate from the extraction pump. It may be supplied directly or via a head tank with automatic level control.

The diaphragms of impulse turbines have labyrinth seals at the shaft. These seals are made of brass of stainless steel and are in six segments, each segment is springloaded and kept against a stop allowing a very small clearance between seal and shaft. In case of a bent shaft, the shaft may push the segment back against the spring pressure, preventing serious damage to shaft or seal ( Fig. 33 )

Turbine Couplings

The purpose of couplings is to transmit power from the prime mover to the driven piece of machinery. For heavy loads the solid flange coupling is used ( Fig. 34 ). The flanges are generally integral parts of the shafts, but they may be separate parts for smaller turbines. In this case each coupling part has a tapered bore and keyway to fit the tapered end of the shaft. Following the taper the shaft has a large thread allowing the coupling to be secured tightly with a large nut.

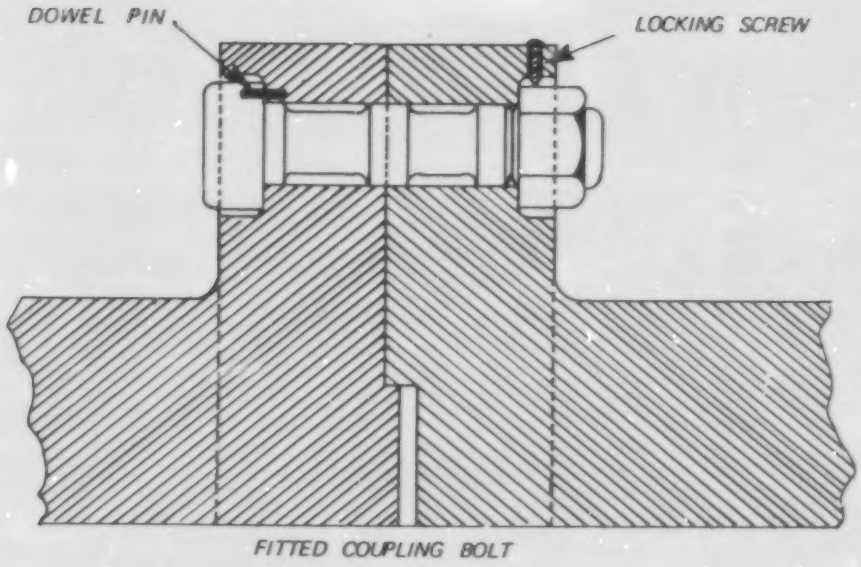
The friction between the coupling halves and the shear force of the bolts transmits the power. For maximum shear stress the bolts must be fitted ( i.e. they must fit in the holes without clearance at the shear point as shown in Fig. 34. The coupling bolts should be undercut, that is machined off to a diameter slightly less than the bottom diameter of the thread to avoid any strain on the thread.

In some cases the couplings must compensate for axial expansion and contraction of the rotors and in this case a flexible coupling is applied ( Fig. 35 ). The outer half has internal gears, while the inner part has matching external gears. The coupling works like the spline on a driveshaft for a car.

The couplings for very large shafts will need a large diameter if the bolts are used to transmit the power. The bolts can be much smaller if they are not allowed to transmit power. In the coupling in Fig. 36 shear pins carry the load. The area exposed to shear is the shear pin diameter x length x number of shear pins. This design allows the shear pins to be located at a large radius from the shaft centre. The coupling bolts are not fitted as they are exposed to tensile stress only.

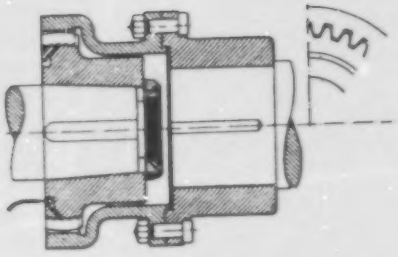
Turbine Blades

The efficiency of the turbine depends more than anything else on the design of the turbine blades. The impulse blades must be designed to convert the kinetic energy of the steam into mechanical energy. The same goes for the reaction blades which furthermore must convert heat energy to kinetic energy. The later years' increase in blade efficiency is due to increased aerodynamic shape calculated by computers and the milling of blades on automatic milling machines.




Solid Flange Coupling with Fitted Bolts

Fig. 34

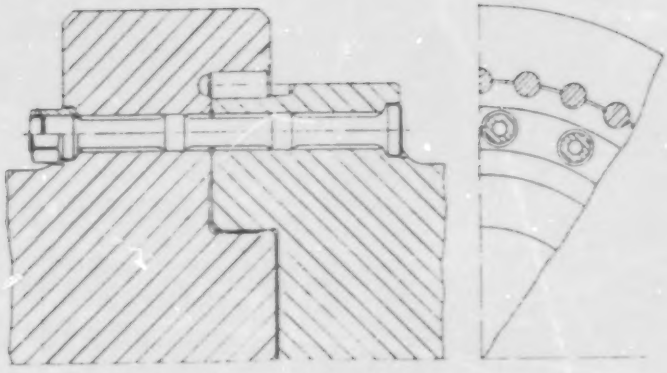


Flexible Gear Coupling

Fig. 35

Shear Pin Coupling

Fig. 36



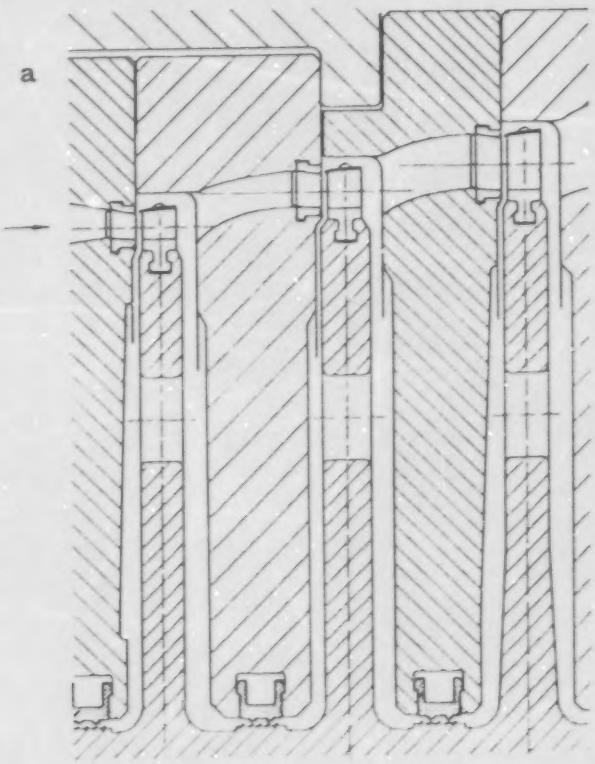
( PE3-4-1-27 )

It is not always possible to give the blades the theoretically best profile, as several other considerations must be taken. The blade must be made strong enough to withstand high temperatures and stresses from heavy, often pulsating steam loads. There is also the stress due to centrifugal force ( for large L.P. blades the centrifugal force on a single blade may exceed 200 tonnes ). Vibrations and resonant vibrations in particular must be taken into account and finally there is erosion and corrosion.

The material that comes closest to the ideal for all mentioned considerations is a chromium-nickel steel, for instance 17 Cr/13 Ni - steel.

Stationary Blades and Nozzles

The first set of nozzles for an impulse turbine is the control set and is divided into three to six sections with each section having a steam control valve. For smaller turbines all sections may be located in the upper half of the casing, while the sections for larger turbines cover the entire circumference. All stages following the control stage have the nozzles located in diaphragms ( Fig. 37 ). The diaphragms are in halves and fitted into grooves in the casing. Locking pieces in the upper part of the casing prevent the diaphragm from turning. All modern diaphragms are of an all-welded construction.



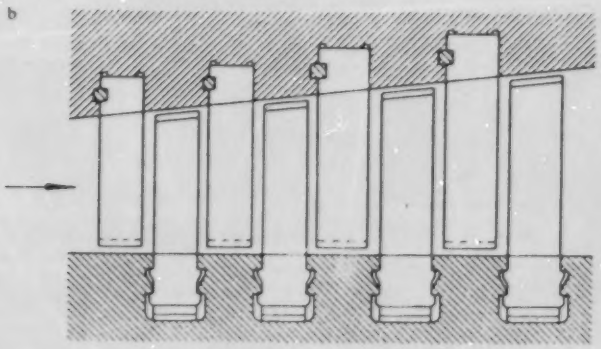

Disc-type Rotor

Fig. 37

( PE3-4-1-28 )

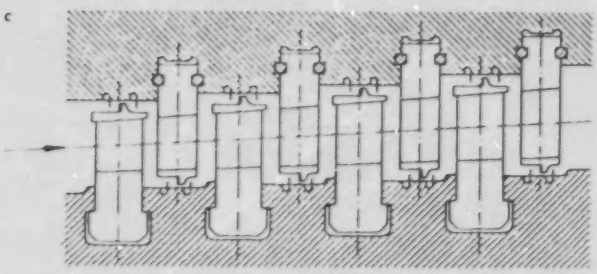
The stationary blades in reaction turbines are fitted into grooves in the casing halves; keys as shown ( Fig. 38 and Fig. 39) lock the blades in place. In some cases the blades have keys or serrations on one side of the root and a calking strip on the other side of the root is used to tighten the blades solidly in the grooves. The blades are often supplied with a shroud band with radial and/or axial sealing strips to minimize leakage losses.

The stationary blades for a Curtis wheel are attached to the casings as are the stationary blades for reaction turbines.



Drum-type rotor with tip-sealed blades

Fig. 38



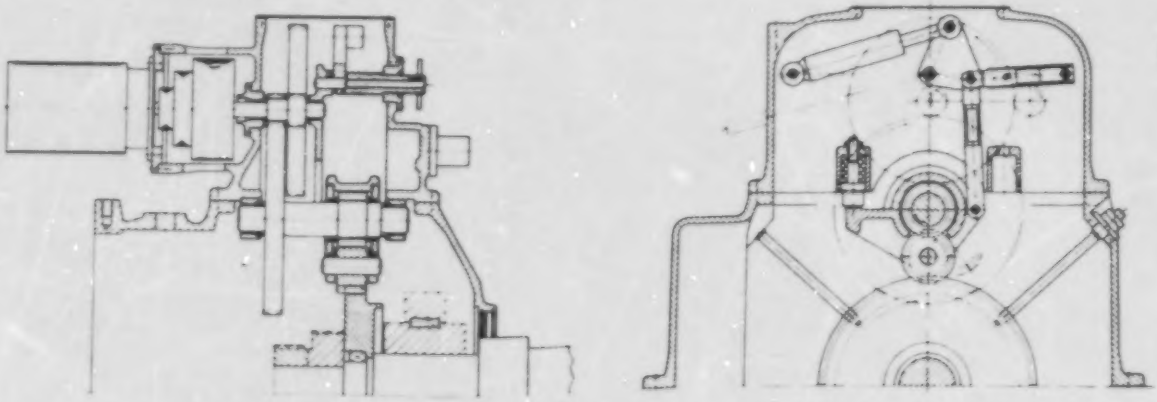
Drum-type rotor with shrouded blades

Fig. 39

Barring Gears

When a turbine is left cold and at standstill the weight of the rotor will tend to bend the rotor slightly. If left at standstill while the turbine is still hot, the lower half of the rotor will cool off faster than the upper half and the rotor will bend upwards "hog". In both cases the turbine would be difficult if not impossible to start up. To overcome the problem the manufacturer supplies the larger turbines with a turning or barring gear consisting of an electric motor which through several sets of reducing gears turns the turbine shaft at low speed. The first turning gears turned the shaft at approximately 20 rev/min later increased to 40 and up to 60 rev/min as proper lubrication is difficult to obtain at low speed; the same goes for the hydrogen seals of generators. Some turning gears, electric or hydraulic, turn the shaft 180° at set times over a period of 24 hours.

Before a cold turbine is started up it should be on the barring gear for approximately three hours. When a turbine is shut down it should be barring for the next 24 hours. If a hydrogen cooled generator is involved the turbine should be kept on barring gear to prevent excessive loss of hydrogen. All barring gears are interlocked with a lubricating oil pressure switch and an engagement limit switch operated by the engagement handle ( Fig. 40 and Fig. 41 ).



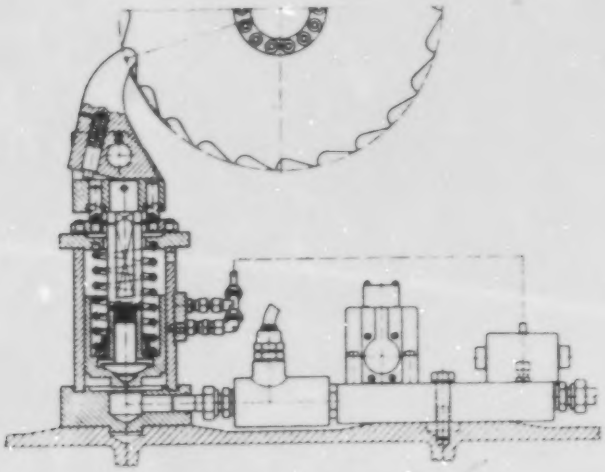
Electric Turning Gear

Fig. 40

Large turbines with heavy rotors are generally equipped with a jacking oil pump, supplying the lower part of the bearings with oil at approximately 10 000 kPa, thereby lifting the shaft and supplying lubricating oil.

( PE3-4-1-30 )

The jacking oil is applied before start-up of the turning gears and for slow rev/min the oil is left on, but for high rev/min ( 50 - 60 rev/min ) it is generally shut down as soon as the turning gear is up to speed.



Hydraulic Turning Gear

Fig. 41

## STEAM TURBINE GOVERNING

### Nozzle Governing

With nozzle governing a series of nozzle valves open in sequence as the load increases. This type of governing is most efficient and is used for impulse turbines ( Fig. 42 ).

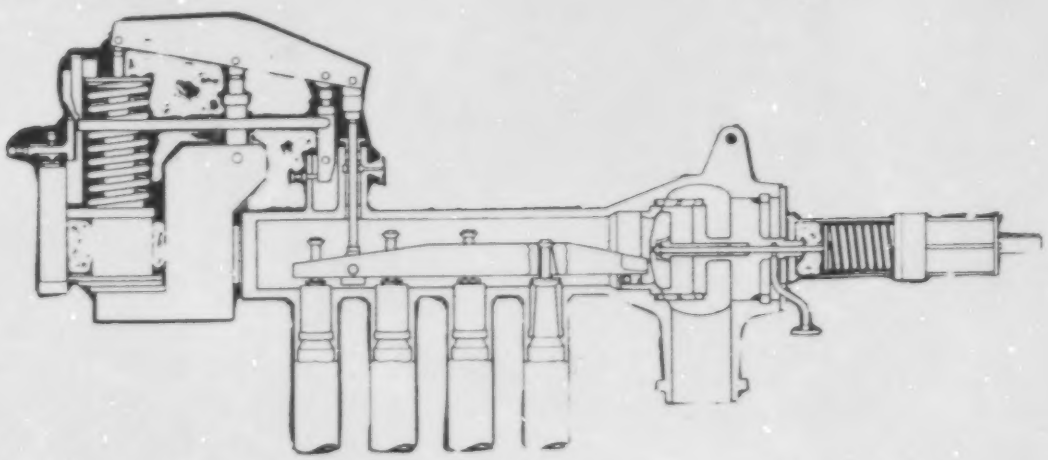


Fig. 42

( PE3-4-1-31 )

Throttle Governing

With throttle governing a single large control valve controls the load from 0% to 100%. For large turbines two control valves operating in parallel replace a large single valve.

When steam is throttled, the superheat increases and the turbine exhaust steam is drier, reducing the turbine blade erosion, but with the drier steam entering the condenser, the condenser losses increase. Throttling of steam through a valve is an isenthalpic ( constant enthalpy ) process and no heat is lost. The so-called throttling losses occur in the condenser.

By-pass Governing or Overload Governing

This system is used on impulse as well as reaction turbines. An extra set of control valves admit steam to the space behind the Curtis wheel or for a reaction turbine to an annular space behind the first 8 - 12 stages. By-passing part of the turbine increases the turbine capacity ( overloads the turbine ) but at a reduced efficiency. The by-pass valves are smaller than the regular governor valves, as too much by-pass steam may starve the by-passed stages rotating in steam at very high density and the blades may overheat.

Turbine Governors

The two general types of governors used are the speed sensitive governor and the pressure sensitive governor. Speed sensitive governors are applied to all kinds of turbines.

Pressure sensitive governors are applied to back pressure and extraction turbines in connection with the speed sensitive governor.

1. Speed Governing

The frequency of 60 Hz is used as the set point or balance between supply and demand of an electric network. Any over supply of energy will increase the frequency and an under supply will decrease the frequency. The supply must at any time be equal to the demand in order to keep the frequency at exactly 60 Hz.

The speed governor is a proportional-action controller, each change in power causes a change in the turbine speed. The governor controls the opening of the control valves as a function of this speed change. On account of the governor speed droop, the frequency is not constant over the full range of load without

The flyballs operate the piston in the governor relay, and thereby controls the leak-off at the right end of the piston.

Assuming the turbine speed increases, the flyballs move outwards and move the relay piston to the right, increasing the leak-off and lowering the pilot oil pressure causing the piston in pilot cylinder H to move downwards. As the piston in cylinder J is stationary, the pilot piston in H will via the linkage move control piston K downwards, opening the oil drain for cylinder J and the piston in J moves downwards, but in doing so the control piston K will be moved upwards as the piston in H is stationary. When K comes back to the neutral position, oil can no longer drain out from J and the governor valve has taken up the position corresponding to the pilot oil pressure.

Turning the handwheel L clockwise will move the relay bushing to the right, decrease the leak-off, increase the pilot oil pressure for further opening of the governor valve and increasing the load. Turning L counterclockwise will reduce the turbine load.

For smaller turbines the pilot oil may be applied directly to cylinder J and K and H are omitted. The valve F is locked in a set position, but if F is opened up more, the increased oil-flow in the pilot oil system makes the system less sensitive, i.e., the speed droop is increased; closing in on F decreases the speed droop.

It has been pointed out that nozzle governing is more efficient than throttle governing and that is one of the reasons for using a velocity compounded impulse wheel as control stage for an impulse reaction turbine. The second reason is that at high pressure the leakage losses around the reaction blades is excessive. Somewhere in one of the boiler lectures it was pointed out how a few extra kJ per kg steam could increase the pressure. When we expand the steam through the turbine we find that at the high pressure a few kJ per kg steam require a large pressure drop. For instance an adiabatic expansion of steam at 12 000 kPa and 500$^{o}$ C to 7000 kPa releases 170 kJ/kg, but the same type of expansion from 40 kPa to 10 kPa also releases 170 kJ/kg, and that is the second reason for the combination of Curtis-Parsons.

Overspeed Trip

The high rotational speed of steam turbines creates large centrifugal forces, as these forces increase with the square of the speed. Therefore an absolute reliable overspeed protection must be provided.

external adjustment. Governor speed droop is the percent change in speed required for a load change equal to the rated capacity of the turbine. It is the same as % proportional band for controllers.

The speed sensitive governor may be:

a ) Mechanical

b ) Mechanical-hydraulic

c ) Electro-hydraulic

A turbine manufacturer may use all three types depending on the size of the turbine; he may furthermore use several systems for each type. It is therefore impossible to describe more than a single example, such as the standard mechanical-hydraulic system applied to most Parsons turbines ( Fig. 43 ).

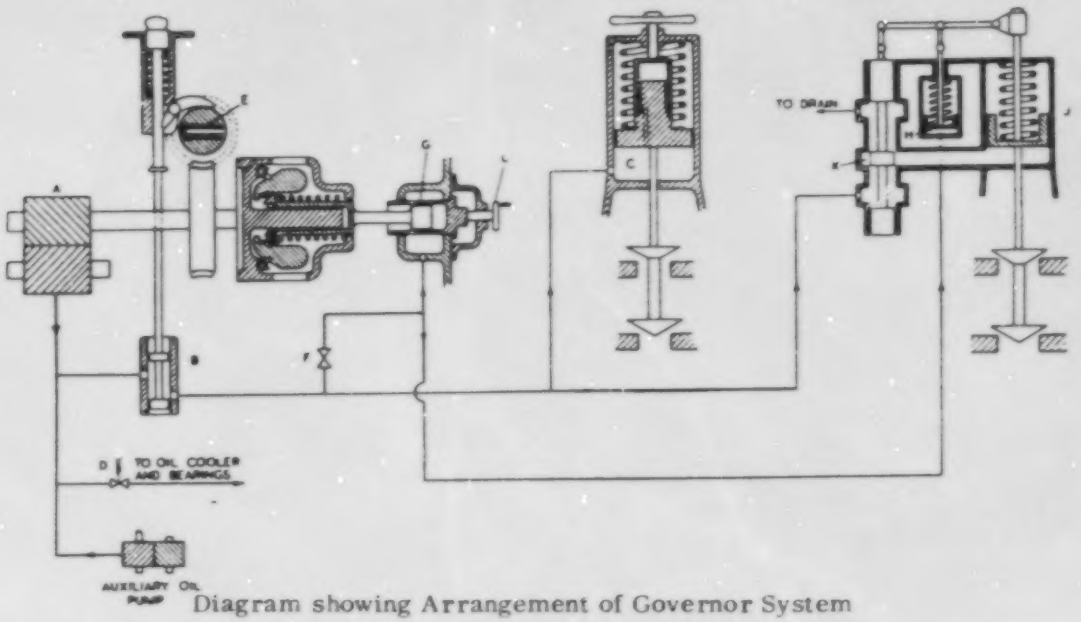



Diagram showing Arrangement of Governor System

Fig. 43

Power oil at approximately 500 kPa is supplied from pump A, through overspeed relay B up to speed droop setting valve F, to main stop valve oil cylinder C and up to control valve K. The oil downstream of F is pilot oil ( 100 - 400 kPa ) up to governor relay G and up to pilot cylinder H.

( PE3-4-1-33 )

The overspeed trip in Fig. 44 is mechanical-hydraulic and shows clearly the operating principle of all overspeed trips for turbines with hydraulic governor systems. The springloaded tripping bolt located in the turbine shaft has the centre of gravity slightly off the centre of the shaft in direction of the bolt head. The nut at the end of the bolt provides a stop for the bolt in tripped position and for the tripping speed adjustment. During normal operation the main spring holds the relay rod against the tripping lever; piston A has closed the oil drain and H. P. oil passes between pistons A and B to the stop valve.

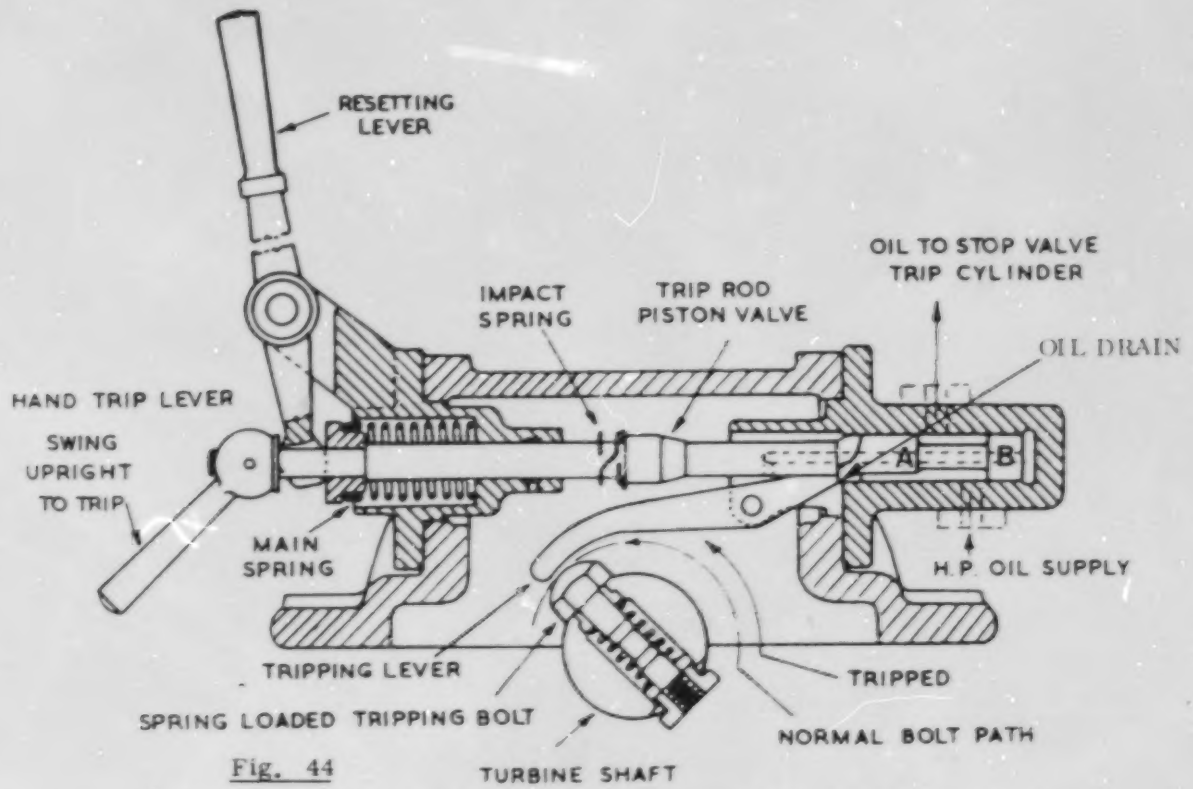



Fig. 44

When turbine speed increases to the trip setting, usually 110% of operating speed, the centrifugal force overcomes the bolt spring, the bolt moves to the tripping position, strikes the tripping lever, unlatching the relay rod and the main spring moves the relay to the tripped position in which piston A opens the stop valve oil-port to drain, while piston B closes off the H. P. oil inlet port.

For all hydraulic systems the overspeed trip closes off for H. P. oil and opens the stop valve oil cylinder to drain allowing the valve to close under spring force.

( PE3-4-1-35 )

- QUESTION SHEET -

POWER ENGINEERING

Third Class
Section 4, Lect. 1

1. Make a blade sketch and the steam pressure and velocity graphs for an impulse turbine with one Curtis and two Rateau stages.

2. Explain the following terms and list their advantages and disadvantages:

   a) Nozzle governing

   b) Throttle governing

   c) By-pass governing

3. a) Explain the working principle of a Kingsbury thrust bearing.

   b) Explain the operating principle of an oil pressure thrust bearing position indicator.

4. The formula for centrifugal force is $\frac{m \times V^2}{r}$

   where $m$ = mass in kg, $V$ = velocity in metre/s of the centre of gravity of blade

   $r$ = radius to centre of gravity of blade

   Using above formula find the centrifugal force in kilonewtons of a turbine blade having a mass of 4.6 kg and rotating at 3600 rev/min when $r$ = 1.5 metres.

5. Sketch and describe an overspeed tripping device.

6. Compare a back-pressure turbine and an extraction-condensing turbine and list their advantages and disadvantages.

7. What is the purpose of a dummy piston and what factors enter into its design?

8. Sketch a solid flange coupling and explain where it would be used.

# 8.2

STEAM TURBINES — COMPONENTS

## Goal:

The apprentice will be able to describe the major components of a steam turbine.

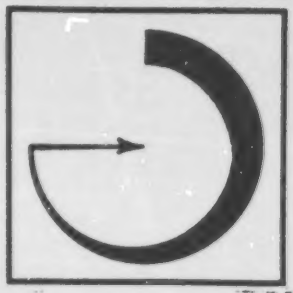
## Performance Indicators:

1. Describe casings, rotors, blading and casing drains.
2. Describe packing glands, governors and extraction valves.
3. Describe speed reduction gears, flexible couplings, and turning gears.
4. Describe lubricating systems, thrust bearings, ring-oiled bearings and pressure-fed bearings.

# Study Guide

* Read the goal and performance indicators to find what is to be learned from package.
* Read the vocabulary list to find new words that will be used in package.
* Read the introduction and information sheets.
* Complete the job sheet.
* Complete self-assessment.
* Complete post-assessment.

# Vocabulary



* Balance pipe
* Blading
* Carbon ring seal
* Casings
* Diaphragm
* Direct connected governor
* Disc rotor
* Dummy piston
* Emergency trip
* End tightening
* Extraction turbine
* Flexible couplings
* Flyweight governor
* Grid type extraction valve
* Hollow drum rotor
* Hydraulic governor
* Impulse blading
* Indirect connected governor
* Labyrinth seal
* Lubricating systems
* Main governor
* Mechanical governor
* Oil circulating system
* Overspeed governor
* Packing gland
* Pressure fed bearings
* Pressure governor
* Reaction blading
* Ring-oiled bearings
* Shroud
* Solid drum rotor
* Solid forged rotor
* Speed governor
* Speed reduction gears
* Tang
* Thrust bearings
* Turning gears
* Water seal
* Welded rotor

# Introduction

A turbine consists of many components and systems for converting steam into mechanical energy. An apprentice must understand how these components function and interact with each other to produce power.

This package provides an explanation of the basic components and systems. With a basic understanding of the turbine, the apprentice will have the framework for learning the technical aspects of turbine operation. The technical details of turbines are too complex to be mastered in a learning package. A package can be a starting point. Experience will bring technical competence with steam turbines.

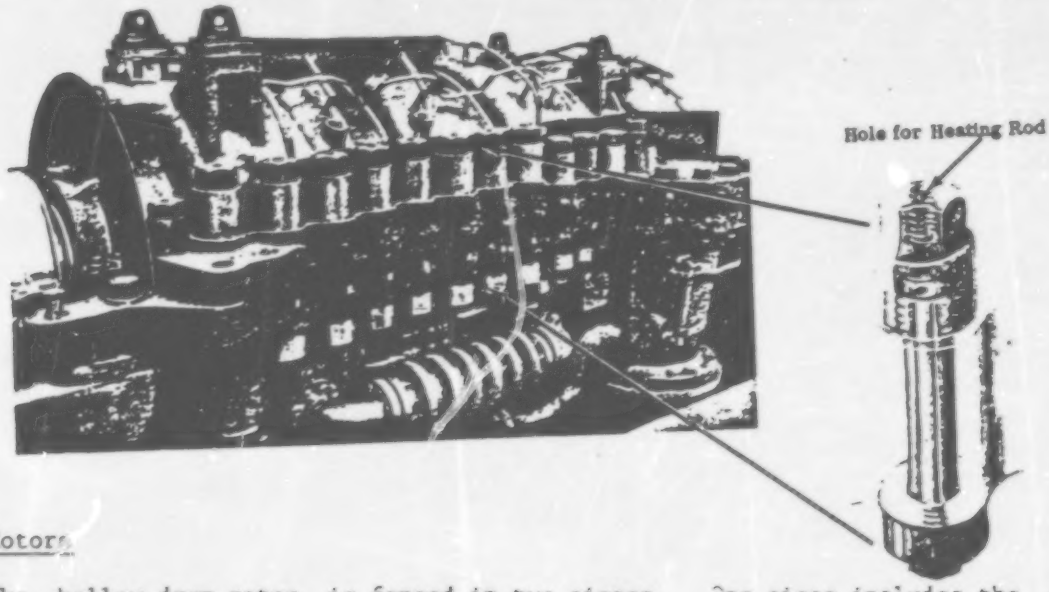
# Information

## Casings

Turbine casings are of simple construction. They are divided horizontally so that one section can be removed for inspection of the turbine. The casing joints are machined smooth to make a close joint. High tensile bolts are used to fasten the two sections of casing together in high pressure casings. A hole has been drilled the length of the bolts to allow for the insertion of carbon heating rods. Heating rods are used so that proper tensioning of the bolts can be made when the bolts are tightened. High pressure casings are made of cast steel.

Low pressure casings are made of cast iron or fabricated steel. The exhaust chambers are braced with plates or stays to avoid distortion of the casing.

A bolted high pressure casing is shown below.

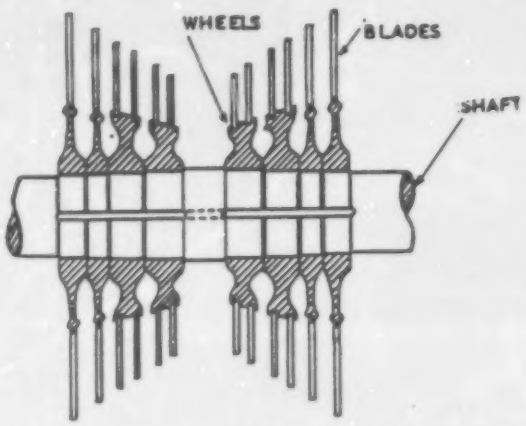



## Rotors

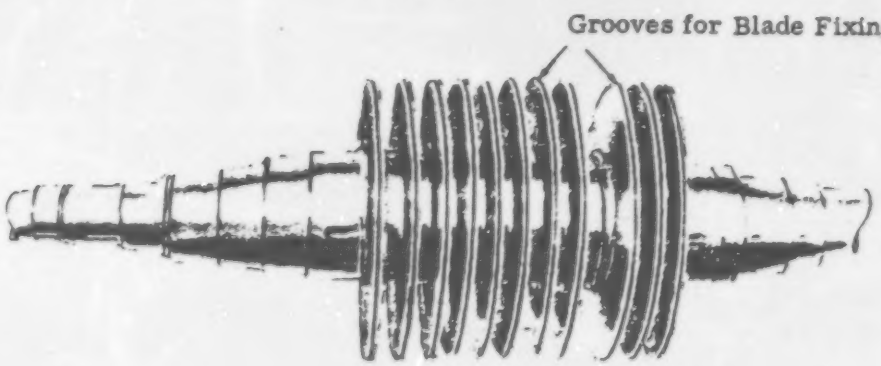
The hollow drum rotor is forged in two pieces. One piece includes the steam inlet and drum. The other piece is the exhaust end shaft and disc. After machining, the drum is shrunk into the exhaust end disc forging and secured by bolts. The hollow drum rotor is limited to small sizes because it is susceptible to stresses. A solid drum is forged as a solid piece. It is useful in large output reaction turbines.

# Information

Disc rotors are made up of discs or wheels which have been forged separately and keyed to a central shaft. The outer rims of the discs are grooved for blades to be attached. A disc type rotor for a low pressure cylinder is shown in the picture below.



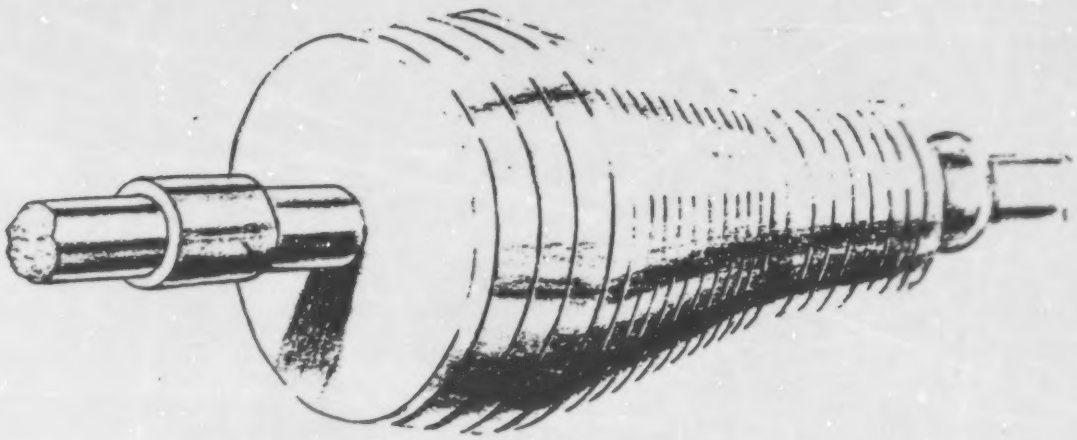

Solid forged rotors have wheels and shafts machined from one piece of metal. Their single piece construction avoids problems of loose wheels. Machined grooves allow blading of the wheels.




Welded Rotors

Welded rotors are made by welding metal discs onto two shaft ends. The discs are then welded together and blading grooves are machined into the outer surfaces. A welded disc is shown on the next page.

# Information



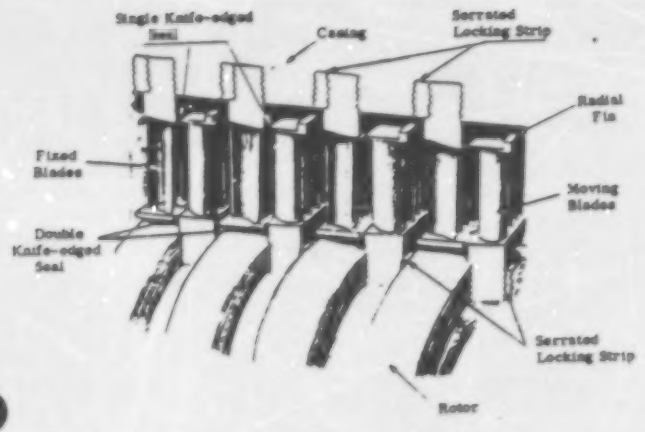
Dummy Pistons

Dummy pistons are machined out of the rotor forging at the steam inlet end. The purpose of adummy piston is to balbance out the force caused by pressure drop as steam passes across each set of blades. A balance pipe helps to maintain the balance of forces obtained by a dummy piston.

Blading

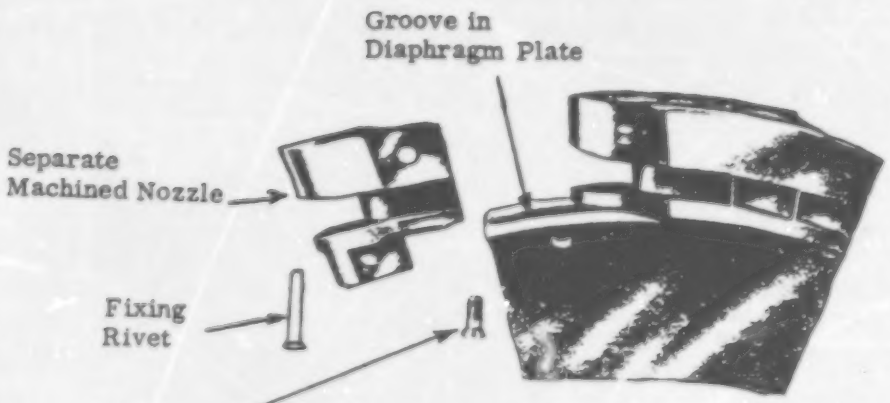
Reaction blading gives a pressure drop across both fixed and moving blades. Fixed blades are fitted in grooves in the casing. Moving blades are fitted into grooves in the rotor. The blades are made complete with shrouding so that installation is easier. The blades are serrated and fit into serrated grooves. They are locked in place by side locking strips. Sealing between the fixed and moving blades is critical for efficient operation. End tightening is a type of sealing provided by controlling the clearance along the line of shaft. Impulse blading uses blades that are machined from a solid bar. A tang isleft at the outer end for attachment of a shroud. The shroud helps to guide steam through the moving blades. The fixed blades of an impulse turbine has nozzles mounted in diaphragms. Diaphragms have two fixed halves attached to the casing. Nozzles are attached in grooves of the diaphragm. The various blade types are shown on the next page.

# Information




H-p Reaction Type Blading

Impulse Type




Built-up Diaphragm

8

# Information

## Cylinder Casing Drains

Water tends to collect at the exhaust end of a turbine. Draining grooves are formed in the casing to allow this water to drain to the condenser. Casing drains keep the water within allowable levels.

## Packing Glands

Turbines tend to leak around the shaft where it emerges from the casing. Air must be sealed out and steam sealed inside the casing. Small turbines use carbon rings that areheld in place by a wire spring. The labyrinth seal is used by larger units. The labyrinth seal consists of a series of rings with sharp projections that extend into grooves on the shaft. The projections are in close tolerance contact with the shaft and prevent steam from passing through them. Some large turbines use a water seal to prevent leakage at the shaft. A ring of water under pressure is maintained on the outer rim of a runner which rotates with the shaft. This seals the gland against leakage. A combination of carbon ring, labyrinth and water seals can be used to reduce leakage at the shaft gland.

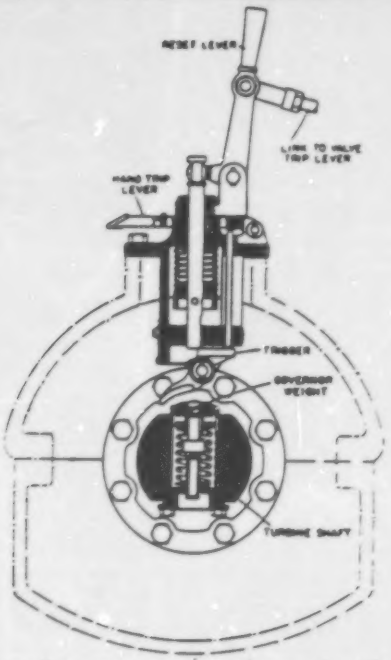
## Governors

Governors control the amount of steam that enters the turbine. Governing is usually controlled by two governors. One is to shut off the steam supply. This is called an overspeed or emergency trip governor. The second one maintains the turbine at a constant speed and is called the main governer. The fly weight governor uses weights on the spindle that revolve with the spindle. Centrifugal force moves the weights outward as the speed is increased. A mechanical linkage connects the governor weights with a valve. When the centrifugal force becomes great enough, the steam valve will be closed. This is a mechanical governor. If the flyweight governor is linked with a hydraulic system, it is called a hydraulic governor. A hydraulic unit is activated by the centrifugal force. The hydraulic system will release oil under pressure to operate a spring-loaded piston. The piston then operates the steam valve.

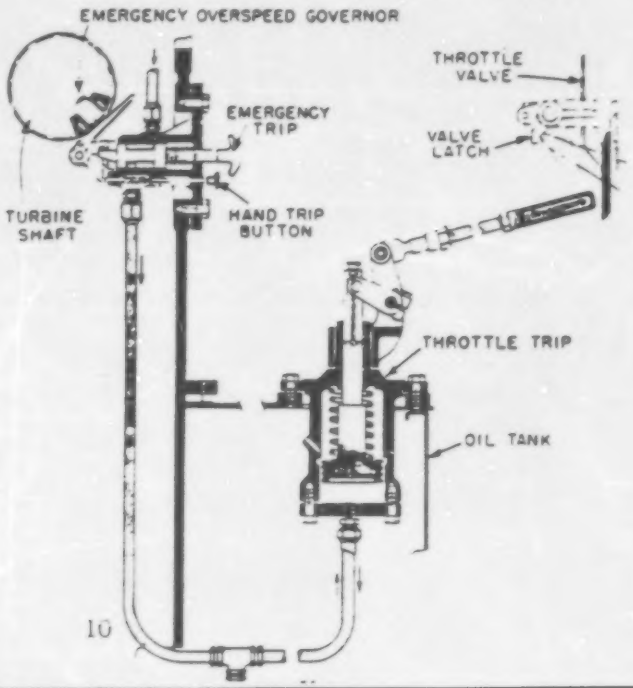
On large turbines, a valve is used to control steam for each set of nozzles. These valves can be operated by bar lift mechanisms, cams or levers or by individual hydraulic cylinders.

A mechanical overspeed governor is shown in the following drawing:

# Information



A hydraulic overspeed governor is shown in the following drawing:




10

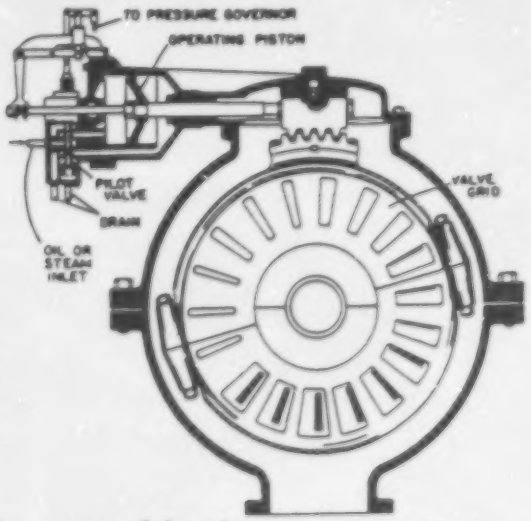
# Information

A governor may be direct connected to the steam control valves through a linkage mechanism. In the application where the governor is linked with a hydraulic unit which trips the steam valves, the unit is an indirect connected governor. All governors consist of three parts:

1. Governor speed sensitive element which is usually flyweight.
2. Linkage which transmits action of the governor to steam control valves.
3. Steam control valves.

Extraction Turbines

Extraction turbines requires a governor that will control the flow of steam beyond the extraction point. Steam must be extracted and yet leave a flow that satisfies steam requirements beyond that point. A grid type extraction valve is made up of a disc that revolves on a grid. Each have ports for steam to pass through. If the ports coincide, a full flow of steam passes through. Partially matched ports allow only a portion of the steam to move by the extraction point. This allows steam to be extracted and still maintain pressure to other parts of the turbine. A grid type extraction valve is shown in this drawing.




The pilot valve is operated by the pressure governor to control the steam to the piston. The piston action rotates the extraction grid. A speed governor is linked with the pressure governor so that the turbine speed is not changed by steam extraction.

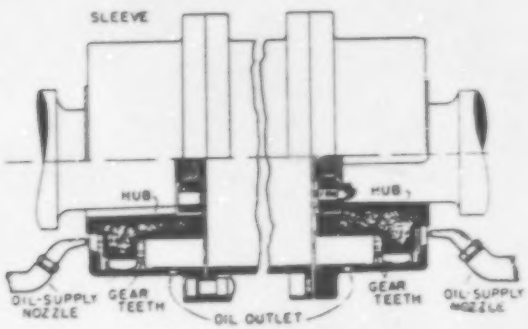
# Information

## Speed Reduction Gears

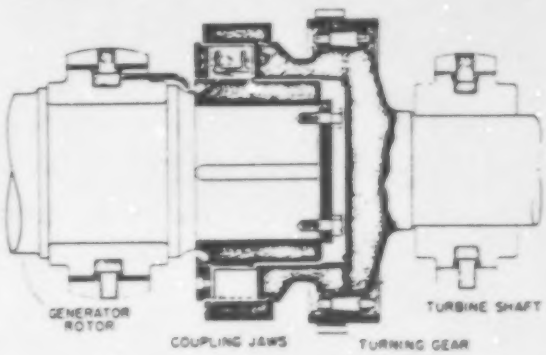
Often the speed of the turbine is too great for the speed of the machine to be driven. Speed reduction gears are used to slow the speed of the turbine. The gear sets are housed in casings and connected to the turbine and unit to be driven by flexible couplings.

## Flexible Couplings

Flexible and claw-type couplings are used to connect the turbine with generators and other driven units.




Flexible Coupling




Claw Type Coupling

## Turning Gears

Turning gears are used to keep the shaft turning after the turbine is shut down. The turning gear is needed to allow the shaft to cool evenly between bearings and avoid distortion from high operating temperatures. It consists of an electric motor and a reduction gear that remains disengaged when not in use. The turning gear is disengaged or engaged with the turbine shaft by use of a yoke and worm gear arrangement.

## Lubricating Systems

Large turbines have oil circulating systems that lubricate bearings, governor mechanisms and generator bearings. Medium size turbines use ring-oiled bearings and an oil circulating system. Small turbines are provided with ring-oiled bearings with some hand oiling of moving parts.

# Information

## Thrust Bearings

Thrust bearings are needed to control axial thrust and maintain position of the moving parts in relation to the stationary parts of the rotor.

## Ring-oiled Bearings

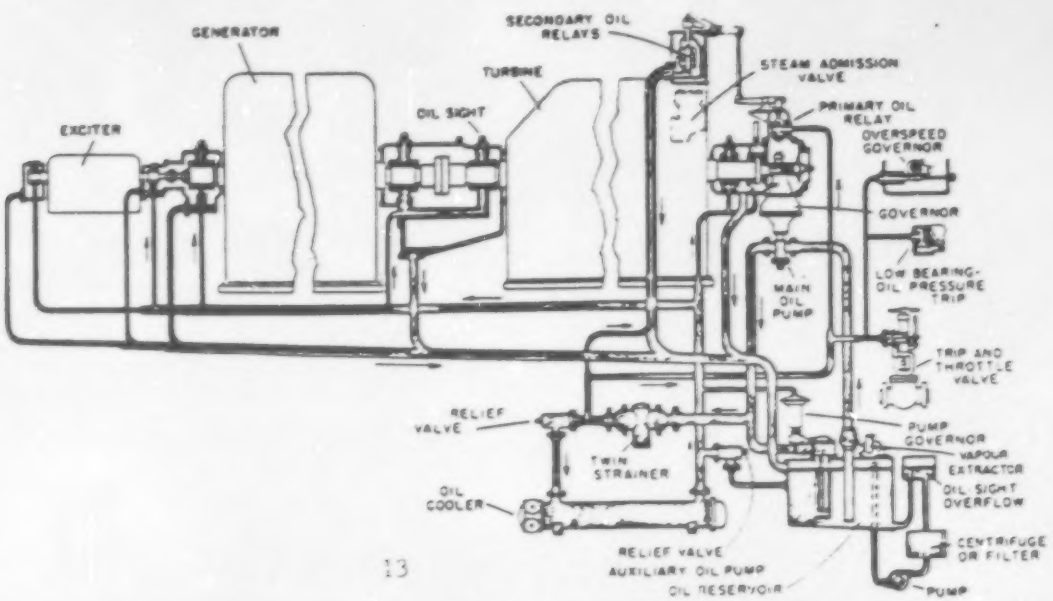
The ring-oiled bearing rides free on the rotating journal of the turbine. As the journal rotates, the ring dips into an oil reservoir and carries oil up to the shaft. Grooves in the bearings channel oil to the bearings.

## Pressure-fed Bearings

The two main bearings of a turbine require a high level of oil to lubricate and control friction. The oil serves as a cooling agent as well as a lubricant for main bearings. To insure an adequate oil supply to the main bearings, a circulating pump is used to deliver oil to the bearings.

## Oil Circulating Systems

In an oil circulating system, oil is delivered at full pump pressure to the governing mechanism. The oil is reduced in pressure and flows to another header that supplies the bearings and other working parts. Oil then returns to the oil reservoir for recycling through the lubrication system.

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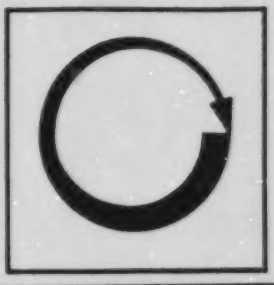
# Assignment



* Read pages 9 - 37 in supplementary reference.
* Complete the job sheet.
* Complete the self-assessment and check your answers with answer sheet.
* Complete the post-assessment and ask the instructor to check your answers.

# Job Sheet

INSPECT A STEAM TURBINE AND IDENTIFY ITS COMPONENTS

* Locate a steam turbine at your job site or other location.
* Carefully inspect the turbine and identify its working components.
* Determine (if possible)
  - How the casing is opened for inspection and location of heating holes in casing bolts.
  - Type of rotor
  - Type of blading
  - Location of casing drains
  - Type of seals in packing glands
  - Type of governoring mechanism
  - Type of turbine (condensing, non-condensing, extraction)
  - If the unit have a speed reduction mechanism
  - How turbine is connected to generator
  - If unit has turning gears
  - Type of lubricating system
* Ask a journeyman to explain components that are encased in housings or not obvious from your inspection.

# Self Assessment

1. High pressure turbine casings are made of _____ _____.

2. _____ rotors are formed by welding metal discs onto two shaft ends.

3. _____ blading of a rotor gives a pressure across both fixed and moving blades.

4. The clearance along the line of shaft can be controlled by a type of sealing called _____ _____.

5. A tang is left at the end of a blade so that the _____ can be attached.

6. List three types of seals used in packing glands.

7. A governor that shuts off the steam supply is _____ governor.

8. A governor that maintains the turbine at a constant speed is the _____ governor.

9. A governor that utilizes a hydraulic unit to trip the steam control valve is called a _____ governor.

10. _____ bearings are used to control axial thrust of the moving parts of a turbine.

# Self Assessment Answers

1. Cast steel
2. Welded
3. Reaction
4. End tightening
5. Shroud
6. Carbon rings, labyrinth seal, water seal
7. Overspeed or emergency trip governor
8. Main governor
9. Hydraulic
10. Thrust

# Post Assessment

1. What type of lubricating system is used on large turbines?

2. What type of lubricating system is used on small turbines?

3. List two types of couplings for linking turbines with generators and other acessories?

4. Turbine speeds can be slowed to the speed of driven machines by the use of _____ _____ gears.

5. List a common type of extraction valve for governing extraction turbines.

6. Which type of governor is used to maintain a constant turbine speed?

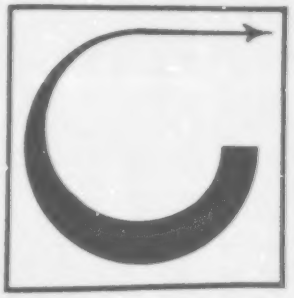
7. Which type of governor is used to shut off the steam supply of a turbine?

8. What is the purpose of cylinder casing drains?

9. What is the purpose of the tang on a blade?

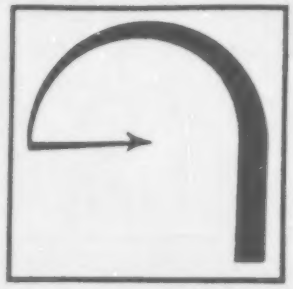
10. What is the purpose of a dummy piston?

# Instructor Post Assessment Answers



1. Oil circulating system
2. Ring-oiled bearings and hand oiling of moving parts
3. Flexible and claw type couplings
4. Speed reduction gears
5. Grid type
6. Main governor
7. Overspeed or emergency trip governor
8. Drain water that collects at the exhaust end of the casing
9. For attachment of the shroud
10. To balance out the force caused by pressure drop across each set of blades.

# Supplementary References



A Correspondence Course. Lecture 4, Section 3. Second class. Southern Alberta Institute of Technology. Calgary, Alberta, Canada.

SOUTHERN ALBERTA INSTITUTE OF TECHNOLOGY
CALGARY, ALBERTA

Correspondence Courses
Power Engineering

SECTION 3

PRIME MOVERS

Second Class
Lecture 4

# STEAM TURBINES 1

## Introduction

It is only during the last fifty years that the steam turbine has really come into its own in the field of prime movers. Modern industry in its quest for super-efficiency and economy dictates that electrical power be produced in central stations and distributed by line to industrial centres and isolated plants alike. This trend has been more than instrumental in the recent great developments which have taken place in steam turbine design and application.

## Principles of Operation

Horizontal steam turbines as installed in stationary plants are produced in a wide range of sizes. The smaller machines, producing up to about 225 kW and running at high speeds, are usually employed to drive exhausters, exciter sets, small lighting generators and other comparatively low power-consuming plants. The larger turbines, some of which are capable of outputs of up to 500 MW, and running at speeds below 4 000 rev/min, are usually found in really large installations such as public power stations and the power houses which furnish power and light for large industrial undertakings.

The revolving member of a steam turbine known as the rotor, consists of a shaft or spindle upon which the blades are attached. The steam in passing through these blades does useful work and so causes the rotor to revolve. This conversion of the potential energy of the steam into useful energy is effected in two stages.

Firstly, the pressure energy is converted into kinetic energy as the steam expands through the nozzles and the pressure drops. These nozzles which may be either stationary or movable are so designed that the steam expands from a high pressure to a lower pressure and in doing so produces the maximum velocity possible in the steam jet.

Secondly the kinetic energy of the jet is converted into useful work by changing the directional momentum of the steam by blades or buckets. In some designs both functions take place simultaneously.

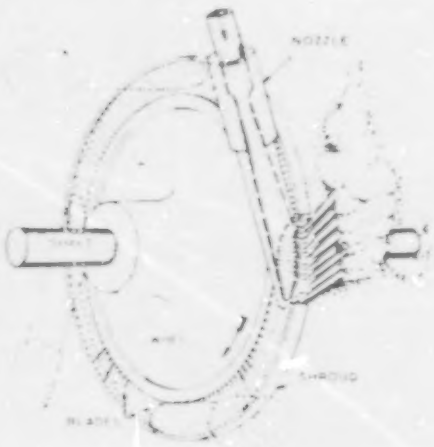
Thus it is obvious that the two most important parts in any steam turbine are the nozzles and the blades, since upon the design and efficacy of these components depends the overall efficiency of the whole machine. Fundamentally the two processes by which the energy in the steam is converted into rotary motion can be broadly divided as follows:

(1) by the impulse type turbine,
(2) by the reaction type turbine.

## THE IMPULSE TURBINE

In the impulse type turbine the steam pressure is reduced in stationary nozzles which also serve to increase its velocity. The high velocity steam impinges upon the blades and imparts an impulse to them rotating the shaft.

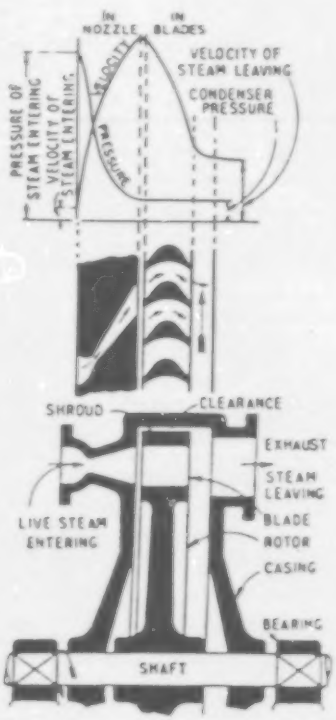
Figs. 1 and 2 show diagrammatically a simple impulse turbine in which the steam is expanded completely in one step. This results in a very high rotational speed.




Rotor and Nozzle of a Simple Impulse Turbine Fig. 1

Steam emerges from the nozzle at a high velocity and strikes the blades which change its direction of flow.

(PE2-3-4-2)

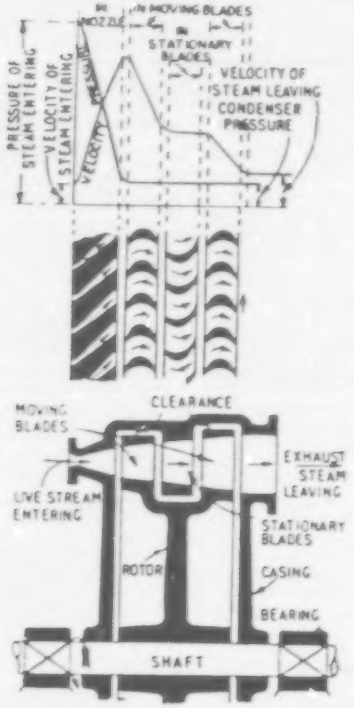



Simple Impulse Turbine Fig. 2

The steam pressure is reduced in the stationary nozzles and the high velocity steam jet impinges upon the blades, thus imparting motion to them.

Lower turbine speeds can be obtained without an excessive loss of efficiency by using two sets of moving blades mounted on the same disc, each absorbing its share of the energy in the steam. This principle is known as Velocity Compounding and is illustrated diagrammatically in Fig. 3.

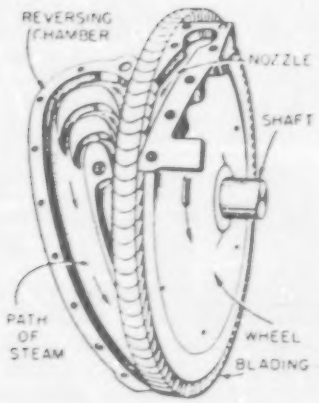
Referring to Fig. 3, steam enters the turbine casing and, after expansion in the nozzle, is directed at high velocity into the first row of moving blades, where part of the velocity is absorbed. After the first row of moving blades, the steam enters a row of stationary baldes which are fixed to the turbine casing and located between the two rows of moving blades. The stationary blades simply change the direction of the flow of the steam so that it will enter the second row of moving blades at the correct angle. By the time the steam leaves the second row of blades all the available velocity has been absorbed.




Velocity Compounded Impulse Turbine
Fig. 3

(PE2-3-1-3)

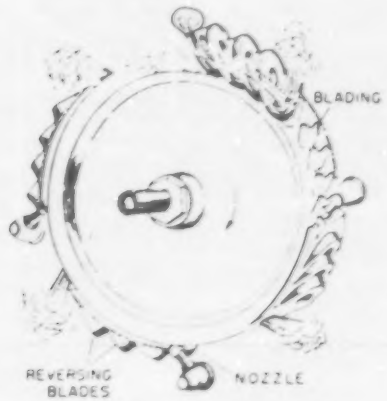
There is another system of velocity compounding in which one set of moving blades is used, known as the re-entry type turbine. In this system the steam, after passing through the blades once, is reversed and directed back again into the same set of blades. Two types of construction make this possible and these are illustrated in Figs. 4 and 5.




Re-entry Method of Velocity Compounding

Fig. 4

Steam issuing from the nozzle at a high velocity passes through the rwo of moving blades to a reversing chamber. In this chamber vanes re-direct the steam back through the same row of moving impulse blades.




Another Method of Re-entry Velocity Compounding

Fig. 5

The steam discharged from the nozzle enters pocket-like buckets on the rim of the wheel. Reversing guide passages placed in the stationary casing opposite the buckets are designed to re-direct the steam back into the buckets again.

In addition to velocity compounding as a method of reducing turbine speeds without loss of efficiency, impulse turbines may be designed to reduce the steam pressure in steps or stages. This is known as Pressure Compounding or pressure staging, each set of expansion nozzles with its impulse blading being known as a pressure stage.

This principle is diagrammatically illustrated in Figs. 6, 7 and 8, and is equivalent to mounting several single-stage impulse turbines on a common shaft. In large units velocity compounding is usually employed in the first pressure stage and pressure compounding in the remaining stages.

(PE2-3-4-4)

Referring to Fig. 6 below, the high-pressure steam expands in the first set of nozzles and then passes through the first row of moving blades. It then expands again in another set of nozzles and passes through another set of moving blades.

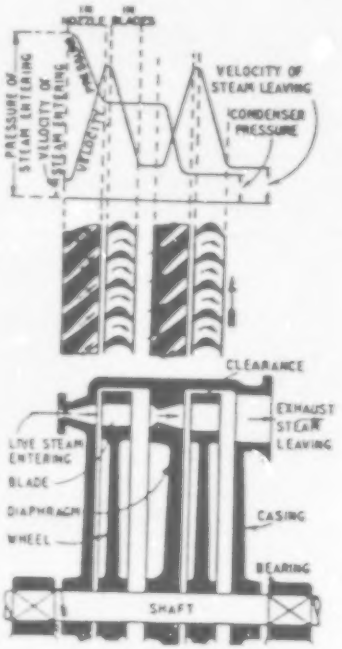
The design shown in Fig. 7 is a combination of velocity compounding and pressure compounding. Each element of the pressure compounding or stage contains a complete example of velocity compounding.

The portion of the turbine shown in Fig. 8 has one velocity and four pressure stages. The graph above the cross-section of the nozzles and blades shows how the pressure and velocity change as the steam passes through the turbine. The pressure drops as the steam expands in the nozzles but remains constant as it passes through the moving blades.

The first stage is designed for a greater pressure drop than the remaining stages and so the velocity of the steam leaving the first nozzle is much higher.

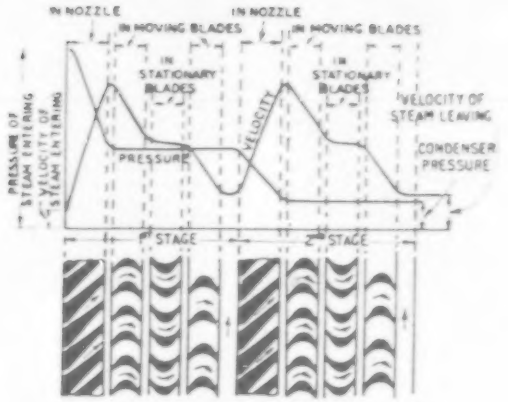
This higher velocity can be used more advantageously to produce work if it is reduced in two steps. Hence, in this stage there are usually two rows of moving blades and an intermediate row of stationary guide blades. The steam velocity decreases in both rows of moving blades and remains constant in the guide vanes.

The diagram is of a pure impulse turbine, but most modern turbines are designed to incorporate a greater or lesser degree of reaction.

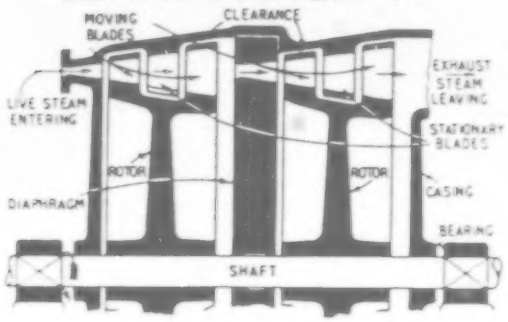


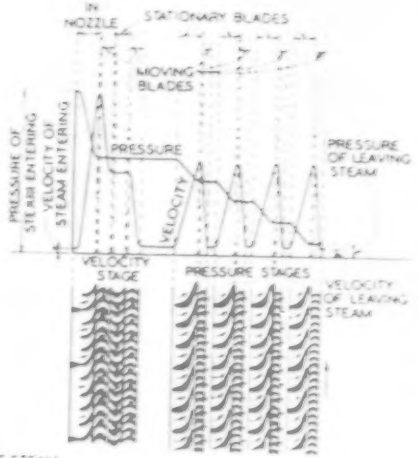

Pressure Compounded
Impulse Turbine
Fig. 6

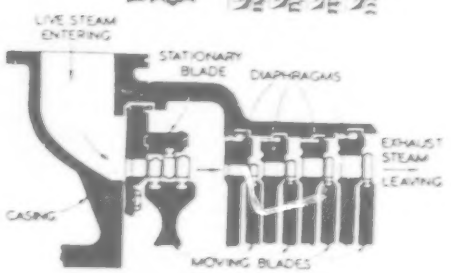
(PE2-3-4-5)




Pressure Velocity
Compounding
Impulse Turbine
Fig. 7




Impulse Turbine with Velocity
and Pressure Stages
Fig. 8

(PE2-3-4-6)

## THE REACTION TURBINE

In the reaction type of turbine, the expansion of the steam occurs in moving nozzle-shaped blades which produce an increase in the velocity of the jets passing through the blades and also a change in direction, as a result of which a driving force is imparted to the blades and so to the rotor.

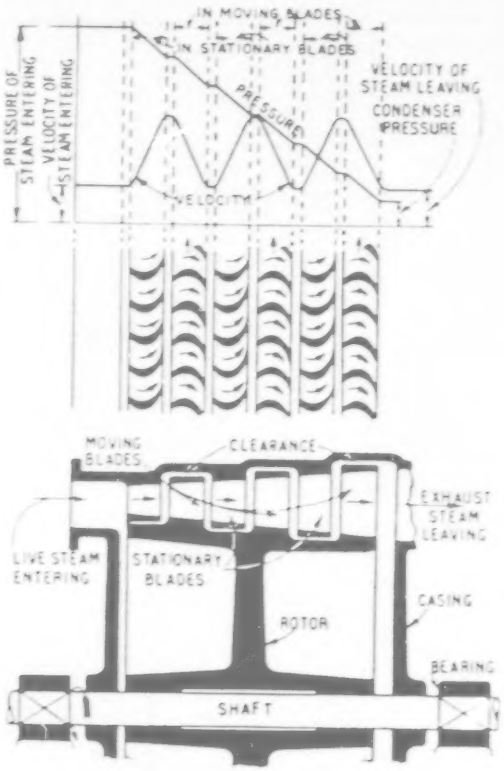
The pure reaction type of turbine is not a practical proposition and is, therefore, never encountered. The introduction of the steam into the last moving blades of the rotor presented some difficulty but the problem was overcome by projecting the steam into the moving blades by means of stationary nozzles.

In theory, the nozzles expand the steam just sufficinetly to give it a velocity equal to that of the moving blades. The steam should then pass into the blades without impact and upon further expansion in the nozzle-shaped blades the potential energy remaining in the steam is converted into kinetic energy.

In actual fact the velocity of the steam produced in the stationary nozzles is greater than the velocity of the blades and the jets strike the latter and produce an impulse effect. Then the further expansion of the steam within the blades produces a reactive force and the rotor is caused to rotate by the the combined action of both the impulse and reactive forces. Turbines acting on this impulse reaction principle are referred to as Reaction Turbines and an illustration of the principle is shown in Fig. 9. Impulse and reaction types of blading are sometimes combined and used together in the same turbine. Fig. 10 shows such an arrangement. The first high-pressure stage is of the velocity compounded impulse type and the low-pressure stages are of the reaction type.

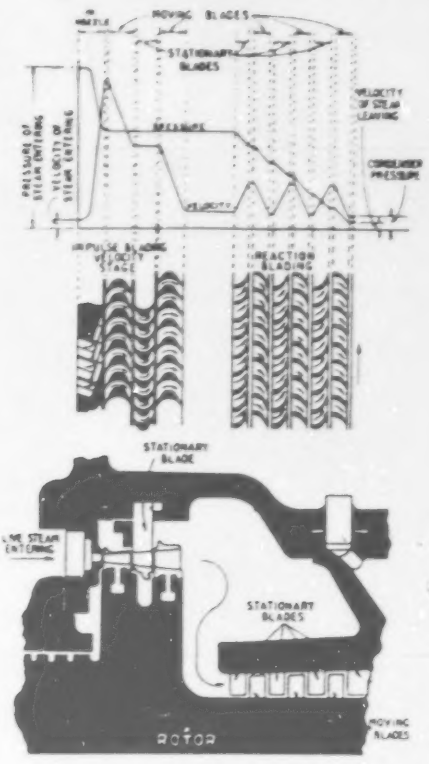
Referring again to Fig. 9, steam under pressure enters a row of stationary blades which direct it into a row of moving blades affixed to the rotor. After leaving this first row of moving blades, the steam enters another row of stationary blades before being passed through other moving blades. The steam pressure drop across both the moving and stationary rows of blades is evident from the graph.

Compound reaction turbines consist of individual high and low pressure units. The steam enters the high-pressure turbine and expands as it flows through the turbine. The low-pressure end takes this steam, expands it still further and exhausts it to a condenser.




Simple Reaction Turbine
Fig. 9

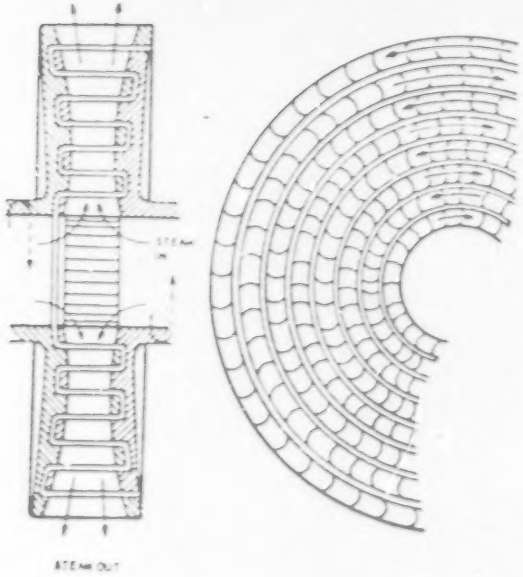
(PE2-3-4-7)




Reaction Turbine with Velocity Compounded Impulse Stage
Fig. 10

With further reference to Fig. 10, in the reaction section of this turbine only three rows of moving and fixed blades are shown. There are, however, many more in the complete turbine. Both the stationary and moving blades form nozzles and the steam pressure drops and velocity rises as it passes through them. Impinging against the moving blades in the direction of rotation the steam imparts energy to them due to impulse.

The moving blades are similar to the stationary blades and the steam expands as it passes through them. This expansion and pressure drop increases the velocity of the steam with respect to the moving blades and, upon leaving, imparts energy to them by reaction. The graph shows the changes in pressure and velocity relative to the stationary blades as steam passes through the turbine. Reaction blades receive more energy from reaction than impulse.



Radial Reaction or Ljungstrom Turbine
Fig. 11

In the radial reaction or Ljungstrom turbine, Fig. 11, the steam flows outward from the centre in a radial direction through reaction nozzles formed by moving blades placed axially. Alternate rows of blades move in opposite directions and as these are connected to two independent shafts these must also revolve in opposite directions. Each shaft is connected to an individual generator.

(E2-3-4-8)

## TURBINE CONSTRUCTION

### Turbine Cylinder Casings

Turbine casings are made as simple in shape as possible so as to reduce to a minimum any distortion due to temperature changes. They are divided into two parts on a horizontal joint so that the machine can be opened up for inspection and overhaul and the rotor removed without disturbing the alignment of the bearings.

Both top and bottom halves of the casing have flanges with holes drilled in them to take fixing bolts. Where there is insufficient clearance for bolts, studs are used.

The joint faces of both top and bottom casings are accurately machined to make a good joint, the pressure on the joint being considerable in the H-p cylinders. High tensile steel bolts are used for the H-p casings, which are usually made of cast steel, and in this case, these bolts have a hole drilled down their length to allow for the insertion of carbon rods for electrical heating when tightening down. This method allows proper tensioning of the bolts to be carried out. Fig. 12 shows a bolted casing joint and an illustration of the type of bolt used.

In some designs of flanges, in order to relieve stresses, vertical saw cuts are made from the edge of the flange through to the bolt holes.

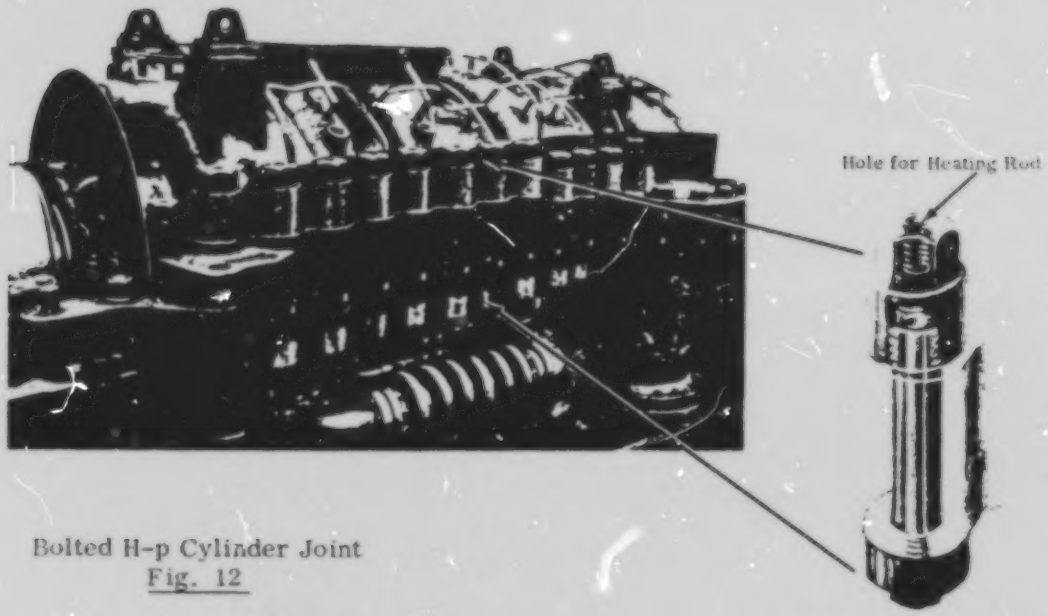
The L-p casing, which is made of cast iron or fabricated steel, is so shaped as to allow adequate passages for the steam as it leaves the last row of blades to flow into the condenser without eddying. The exhaust chambers, which are connected to the condensers, are braced with plates and stays to prevent distortion or collapse of the casing.

The condensers are placed below the L-p casing and the mass is taken through springs supported from the basement-level floor. The mass of the turbo-alternator is carried by the steel and concrete foundation block.

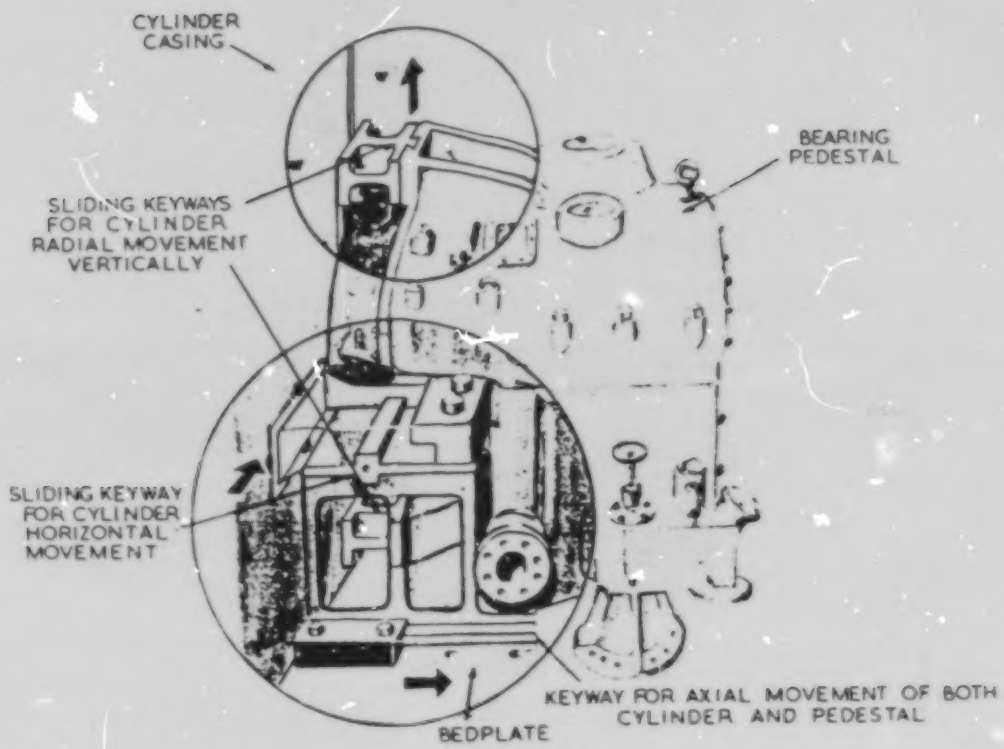
To ensure that correct alignment of the turbine is maintained under all conditions of operation, provision is made to allow free axial expansion and radial expansion. When referring to turbine steam flow, or expansion or contraction, the term axial means along the line of the shafts, and radial (or transverse) means at right angles to or across the line of the shafts. The L-p cylinder casing exhaust is usually anchored to the foundation axially only, at one point, and movement of both the L-p and the H-p cylinder casings is allowed to take place, by means of sliding supports or keyways, to suite the particular design of turbine.

An example of these sliding supports for a H-p cylinder is shown in Fig. 13(a). It will be seen that the cylinder casing is rigidly connected to the bearing pedestal and yet is free to move radially out from the shaft in all directions and still remain in alignment. The bearing pedestal is allowed to slide axially on keyways fitted between the stool or bedplate and the pedestal. An example of an arrangement for the expansion of a two-cylinder turbine is shown in Fig. 13(b).

(PE2-3-4-9)

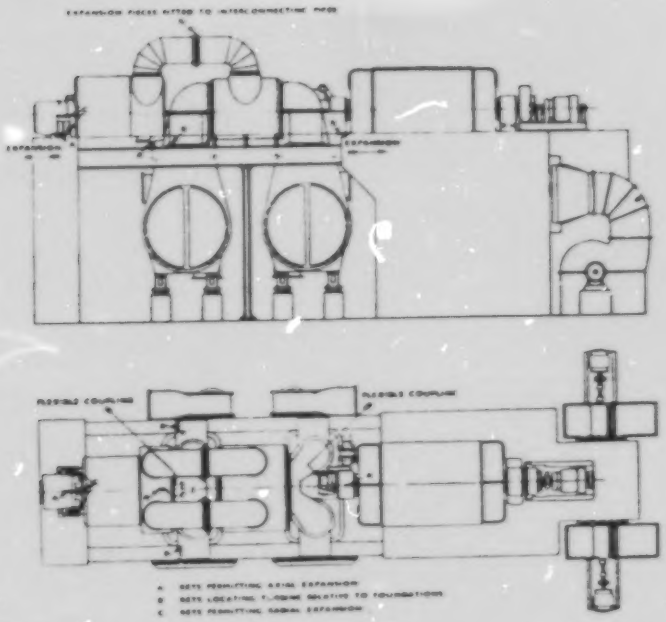



Bolted H-p Cylinder Joint
Fig. 12


CYLINDER CASING
BEARING PEDESTAL
SLIDING KEYWAYS FOR CYLINDER RADIAL MOVEMENT VERTICALLY
SLIDING KEYWAY FOR CYLINDER HORIZONTAL MOVEMENT
KEYWAY FOR AXIAL MOVEMENT OF BOTH CYLINDER AND PEDESTAL
BEDPLATE

H-p Pedestal and Casing Support
Fig. 13(a)

(PE2-3-4-10)



Provision for Expansion of Two-cylinder Turbine
Fig. 13(b)

In some turbines the pedestal bearings are fixed solid to the foundation and the casings are allowed to expand axially at one end by means of supporting feet or "paws", as they are often called, and sliding keyways. Radial expansion keyways are used similar to those described for Fig. 13(a).

Turbine Rotors

Turbine rotors may be of four main types as follows:

1. Forged Steel Drum Rotor
2. Disc Rotor
3. Solid Forged Rotor
4. Welded Rotor



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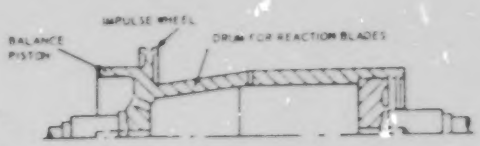
Drum Rotors

In the hollow drum type rotor, Fig. 14(a) the H-p steam inlet end of the rotor and drum is a single steel forging and the exhaust end shaft and disc is another separate forging. After machining, the drum is shrunk onto the exhaust end disc forging and secured by bolts and driving dowels. Grooves are machined in the body of the drum to take the necessary blades.

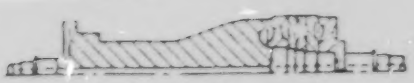
The hollow drum type rotor is limited in its application because of the excessive stresses which would occur if it were made in large sizes. The main advantage of this construction is that there is approximately the same mass of metal in the rotor as in the cylinder casing. Therefore, there is the same response to changing temperature conditions in both rotor and casing and working clearances can be kept to a minimum.

Drum rotors are used to carry the reaction type blading in the H-p cylinders of the Parsons design of turbine. Fig. 14(a) illustrates the construction of the hollow drum type rotor.

The solid drum type rotor, Fig. 14(b) is suitable for cylinders where there are lower temperatures but large diameters. For example they are used in the intermediate cylinders of large output reaction turbines.




Hollow Drum
(a)



Solid Drum
(b)
Fig. 14

(PE2-3-4-12)

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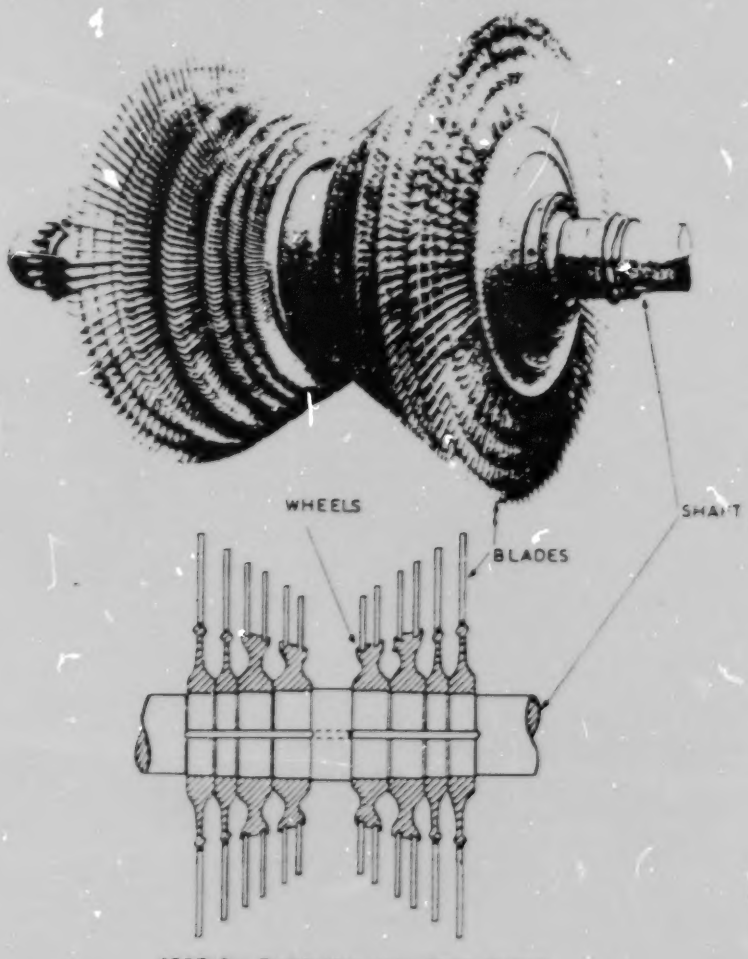
Disc Rotors

The disc rotor is made up of a number of separately-forged discs or wheels and the hubs of these wheels are shrunk or keyed onto the central shaft. The outer rims of the wheels have suitable grooves machined to allow for fixing the blades. The shaft is sometimes stepped so that the wheel hubs can be threaded along to their correct positions. Suitable clearances are left between the hubs to allow for expansion axially along the line of the shaft.

Under operating conditions the temperature of the wheels may rise quicker than that of the shaft and this might tend to make the wheel hubs become loose. To avoid any such danger considerable care is taken during construction of the rotor to ensure that the wheels are shrunk on tight and correctly stressed. Fig. 15 illustrates a disc type of rotor which is the type used in the L-p cylinder of most designs of large turbines.




Disc Type Rotor
Fig. 15

(PE2-3-4-13)

Solid Forged Rotors

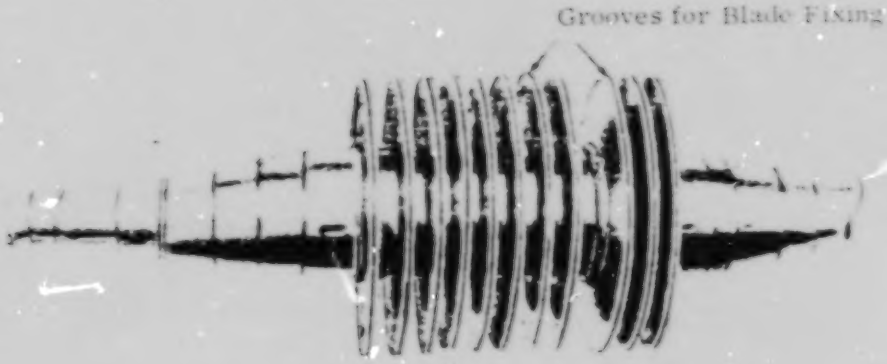
Rotors of this type have wheels and shaft machined from one solid forging, the whole rotor being one complete piece of metal. This results in a rigid construction, and troubles due to loose wheels of the shrunk-on type are eliminated. Grooves are machined in the wheel rims to take the necessary blading.

Solid forged rotors are used in the H-p and I-p cylinders for most designs employing impulse type blading and for the I-p cylinder when reaction type blading is used. Fig. 16 shows a rotor of the solid forged type.

Welded Rotors

Welded rotors are built up from a number of discs and two shaft ends. These are joined together by welding at the circumferences and because there are no central holes in the discs the whole structure has considerable strength. Small holes are drilled in the discs to allow steam to enter inside the rotor body to give uniform heating when coming on load. Grooves are machined in the discs to carry the blades and Fig. 17 (a) and (b) show this type of rotor construction.

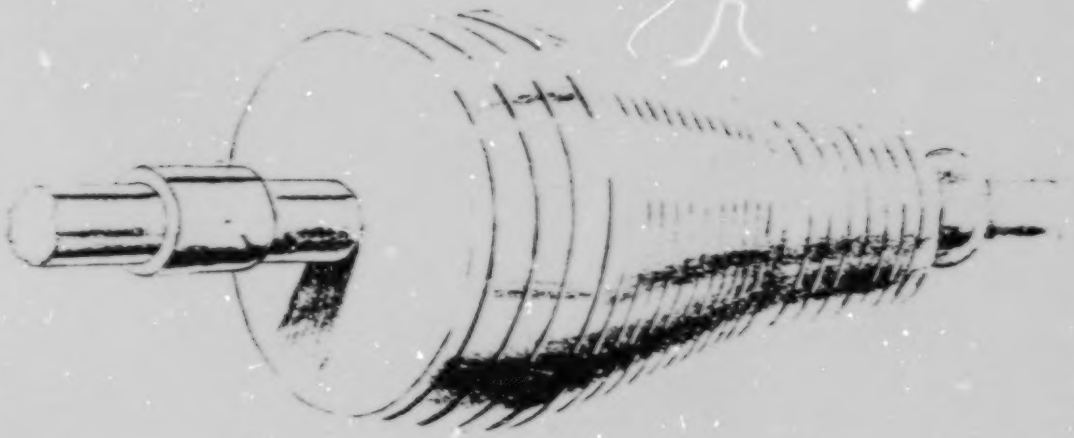
Welded rotors are used in the Swiss Brown-Boveri designs of turbines in the I-p and L-p cylinders.




Solid Forged Rotor
Fig. 16

(PE.2-3-4-14)

Rotor showing Discs before Welding
(a)



Rotor after Welding Discs together
(b)

Fig. 17

(PE2-3-4-15)

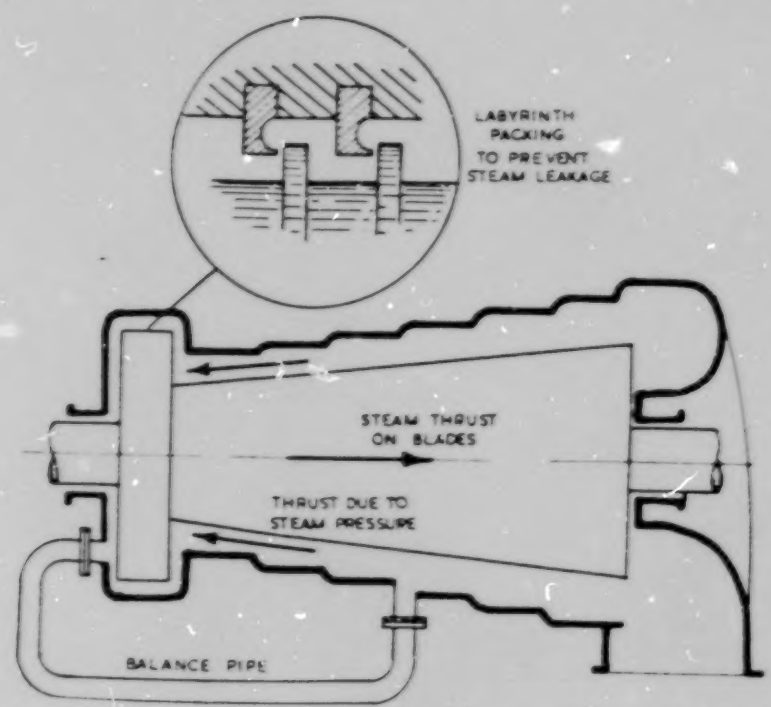
Testing of Rotors

Great care is taken during the manufacture of turbine rotors because they must be sound and as free from internal stresses as possible. Many tests are carred out to ensure that any flaws in the metal are detected before final dispatch of the rotor to site for erection in the turbine.

Dummy Pistons

There is a pressure drop across each row of blades in a reaction type turbine and a considerable force is set up which acts on the rotor in the direction of the steam flow. In order to counteract this force and reduce the load on the thrust bearings, dummy pistons are machined out of the rotor forging at the steam inlet end. It is usual to have a dummy piston for each cylinder except for the L-p double-flow cylinders where the steam flows in both directions and the forces are balanced out.

The dummy piston diameter is so calculated that the steam pressure acting upon it in the opposite direction to the steam flow, balances out the force on the rotor blades in the direction of the steam flow. It is preferable that the dimensions be so arranged as to keep a small out definite thrust towards the exhaust end of the turbine. To help maintain this condition at all loads a balance pipe is usually connected from the casing, on the outer side of the balance piston, to some tap-off point down the cylinder. Both dummy piston and balance pipe arrangement are shown in Fig. 18.




Dummy Piston and Balance Pipe
Fig. 18

(PE2-3-4-16)

Blading

No detail in the construction of a turbine affects the reliability and efficiency more than the design of the blading. This applies particularly to the moving blades which are attached to the turbine rotors.

Depending upon the design of the turbine there is either an impulse force (impulse type blading) or a combination of impulse and reaction forces (reaction type blading) acting on the turbine blades due to the steam flow. The longer the blade the greater the bending force at the root or fixing point of the blade.

In addition there is a centrifugal force, due to the speed at which the blade is rotating, trying to throw the blade outwards.

These two forces - the bending force and throwing-out force - are at maximum in the largest blade wheel at the L-p exhaust end of the turbine. Thus the stresses which these forces impose, limit the size of the blades and the diameter of the last wheel. This limitation is one of the reasons why turbines are designed with double flow in the L-p cylinder.

The mechanical stresses just described are not a great problem in the small H-p moving blading but this blading is subject to much higher temperatures and becomes the greater problem from the designed aspect.

The fixed blades are secured to the turbine cylinder casings and in the impulse type of blading take the form of nozzles set in diaphragms alternately between each row of moving blades.

Reaction Type Blading

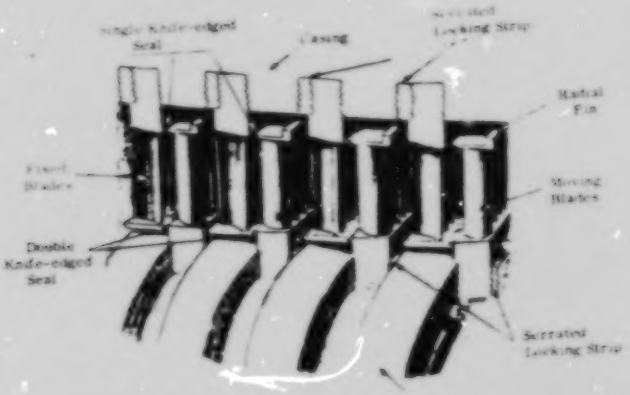
In reaction type blading a pressure drop occurs across both the fixed and moving blades. In the H-p cylinder a very effective seal between fixed and moving blading is essential to prevent steam leakage which would make the turbine inefficient.

The fixed blades are fitted in grooves in the cylinder casing and the moving blades in grooves machined in the rotor.

For blading subject to high temperature in H-p cylinders, the blades are made complete with root section and shrouding in one piece and are formed in groups or packets for convenience of handling. The shrouds have a projecting portion which is thinned down to form a single knife edge on the moving blades. On the fixed blades a second strip is added which is tapered to form a double knife edge. The blade packets are then fitted in the grooves to form a complete row of either fixed or moving blades. The blade packets are serrated along the roots and secured in the grooves which are also serrated, by means of side-locking strips.

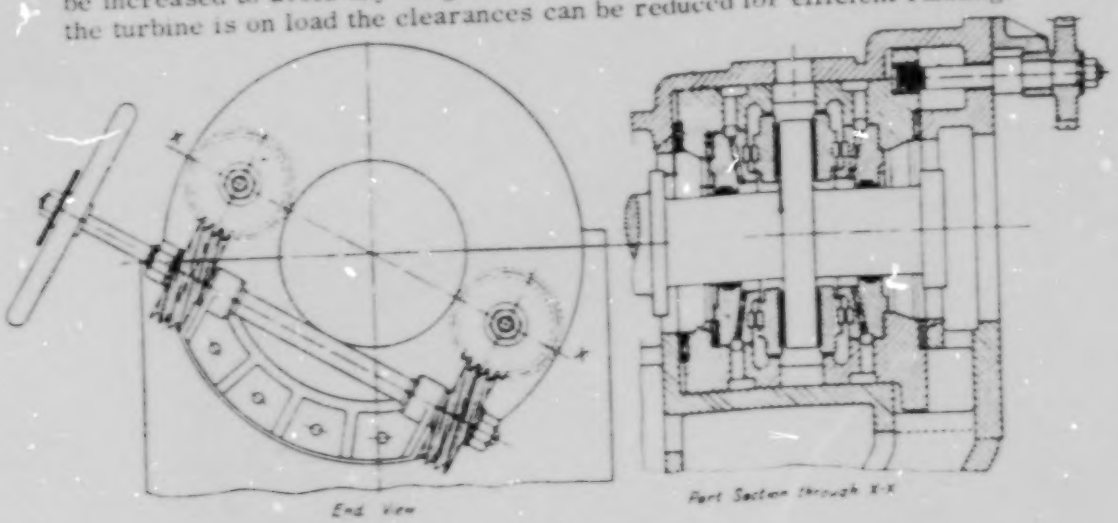
An illustration of this type of blading is shown in Fig. 19, and it will be seen that the leakage of steam is controlled by the axial clearance, that is, the clearance along the line of the shaft. This type of sealing is known as "end tightening".

An additional seal is provided by a radial fin machined into the shroud and set at a reasonably fine clearance between cylinder bore or rotor body.




H-p Reaction Type Blading showing End Tightening
Fig. 19

Because of the very fine clearances which are necessary with this type of blading, thrust adjusting gear is fitted, Fig. 20. This enables the axial position of the rotor to be controlled within strictly defined limits. When the machine is running up, the clearances between fixed and moving blading can be increased to avoid any danger of rubbing due to uneven temperatures. When the turbine is on load the clearances can be reduced for efficient running.




Turbine Thrust Adjusting Gear
Fig. 20

(PE2-3-4-18)

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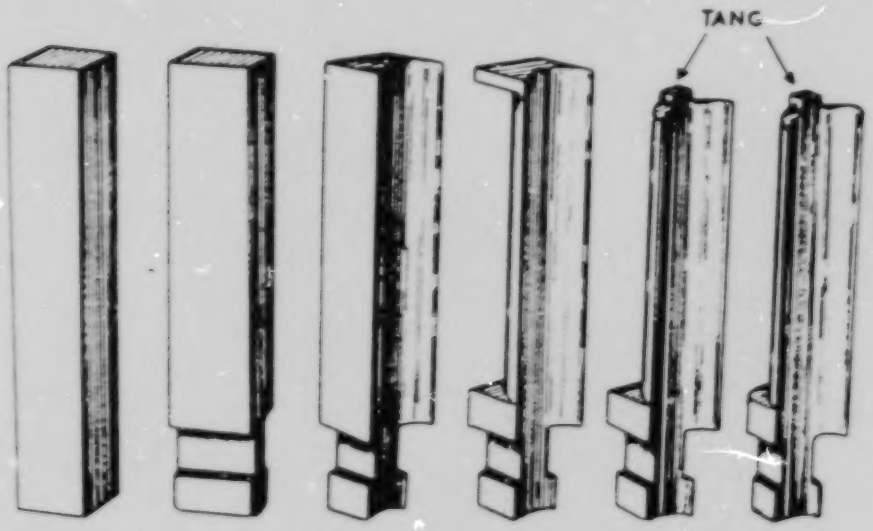
Impulse Type Blading

The H-p moving blades for impulse type turbines are machined from solid bar and the roots and spacers formed with the blade. This is illustrated in Fig. 21. Such construction avoids the use of distance pieces or packers when assembling the blades in the wheels. Tangs are left at the tips of the blades so that when fitted in position in the wheel the shrouding can be attached. The shrouding is made up from sections of metal strip punched with holes to correspond with the tangs. The strip is passed over the tangs which are then splayed out to secure the strip in position. The shrouding is fitted in separate sections to allow for expansion.

There is no pressure drop across the moving blades of an impulse type turbine and therefore the sealing arrangements are not of such great importance as in the reaction type. The shrouding on the impulse blading helps to guide the steam through the moving blades, allowing larger radial clearance, as well as strengthening the assembly.

The fixed blading in an impulse turbine takes the form of nozzles mounted in diaphragms. The diaphragm is made in two halves, one half being fixed to the upper half of the cylinder casing and the other half diaphragm to the lower half cylinder casing. The diaphragms are located in the cylinder casings by means of keys so that when expansion occurs, fouling of the shaft seals is avoided. Special carrier rings are generally used to support the diaphragms in H-p cylinders.

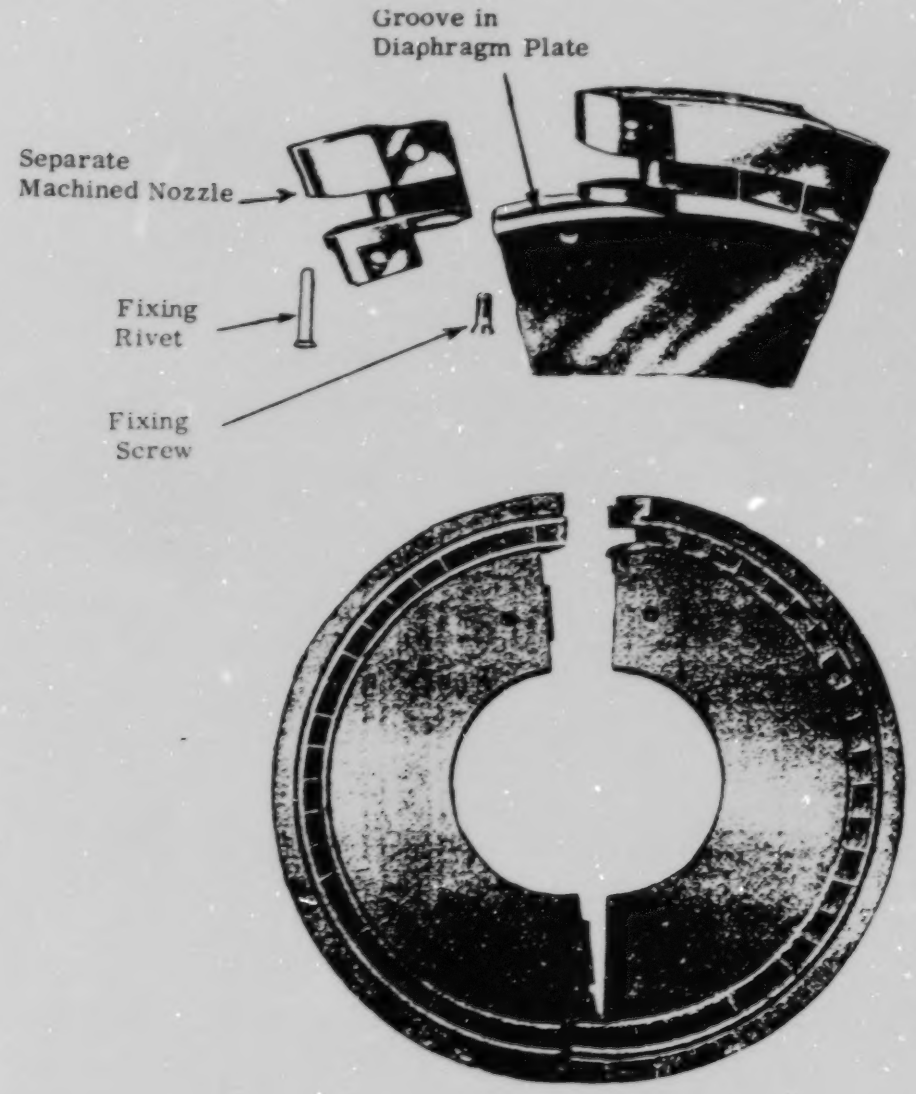



Stages in the manufacture of H-p Impulse Type Moving Blades
Fig. 21

(PE2-3-4-19)

At the H-p end of the turbine the diaphragms are of the built-up type. Each nozzle is machined separately from a solid bar and attached by grooves and rivets to the diaphragm plate. In some cases the nozzles are also welded together and to the plate. Fig. 22 shows the manner in which the nozzles are built up around the diaphragm plate.

Due to the steam pressure difference on each side of the diaphragm it is necessary to provide seals at the hole, where the shaft passes through the diaphragm, to prevent steam leakage along the shaft.




Built-up Diaphragm
Fig. 22

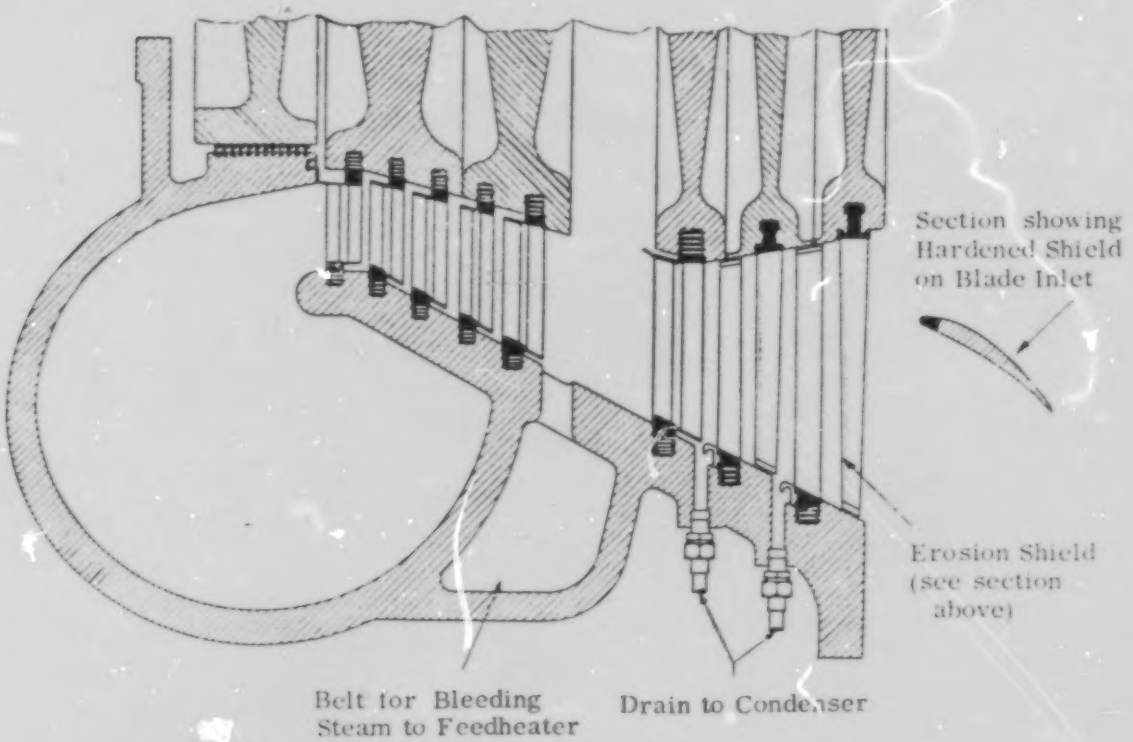
(PE2-3-4-20)

Cylinder Casing Drains

Increased steam pressure and the higher efficiencies of modern turbines have increased the percentage of wetness at the exhaust end; 11% is generally taken as the maximum allowable. The shape of the cylinder casing allows this water to drain to the condenser but special draining grooves are arranged in the cylinder casing to help remove this water more effectively.

An example of this type of draining arrangement is illustrated in Fig. 23.

The L-p blades have to operate at high speed in wet steam and particles of water can cause severe erosion (or wearing away) of the blades. To overcome this difficulty and give blades a longer life a very hard wearing layer or shield of Stellite material is deposited on the steam-inlet edge of the blades. A section through a blade so treated is also shown in Fig. 23.


Section showing Hardened Shield on Blade Inlet
Erosion Shield (see section above)
Belt for Bleeding Steam to Feedheater
Drain to Condenser

Cylinder Casing Drainage
Fig. 23

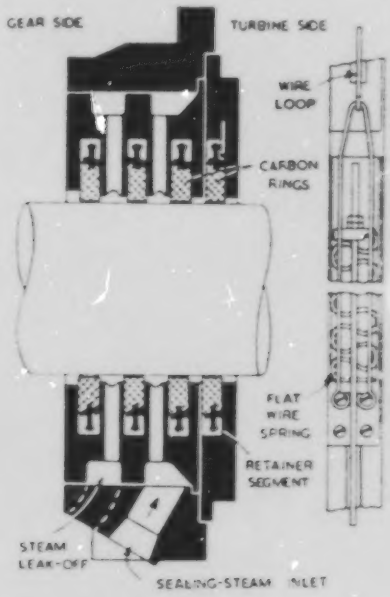
(PE2-3-4-21)

Packing Glands

To prevent the leakage of steam out or of air in around the turbine shaft where it extends through the casing, some form of seal or gland is necessary. When the internal steam pressure is above atmospheric there will be a tendency for the steam to leak out along the shaft past the seals, and when the internal steam pressure is below atmospheric the tendency is for air to leak along the shaft in the opposite direction and reduce the vacuum to which the turbine exhausts.

There are three common types of gland at present in use - carbon rings, labyrinth seals and water seals. Small turbines usually employ the first mentioned although large units may use either of the latter or a combination of the two.

Carbon ring seals, Fig. 24, consist of a number of carbon rings which fit in annular slots in a horizontally-split housing. Each ring is composed of several segments and these are clamped together around the shaft by means of a spring. When properly adjusted the rings should fit snugly onto the shaft but should not grip it. Steam is admitted between the second and third carbon rings to act as an additional seal to prevent the ingress of air when the pressure in the turbine is below atmospheric. A steam leak-off point is provided between the last two rings.

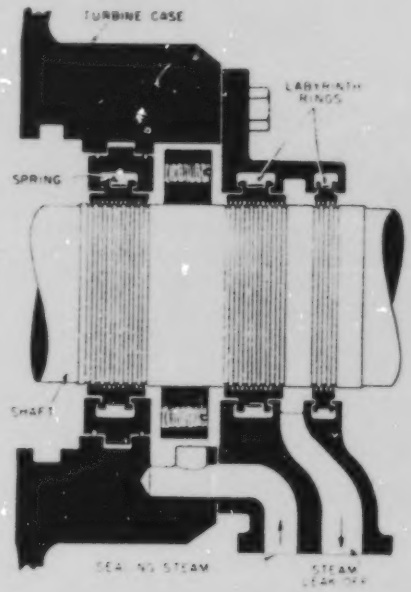



Carbon Ring Shaft Seal
Fig. 24

(PE2-3-4-22)

The labyrinth seal is composed of a number of rings on the inner circumference of which are formed a series of sharp projections, Fig. 25. These rings are mounted in a casing around the turbine rotor shaft and are normally maintained in position by means of a spring. The sharp projections usually fall into grooves either in the shaft itself or in a sleeve fitted over the shaft, although in some designs the shaft in the seal area may be smooth.

The clearances between the projections and the shaft are exceedingly fine and thus a formidable labyrinth is formed around through which any leakage of steam or air must flow. This series of small circular spaces in the labyrinth offers a great resistance to steam flow and causes a series of pressure drops along the shaft which minimize effectively any tendency to leakage.

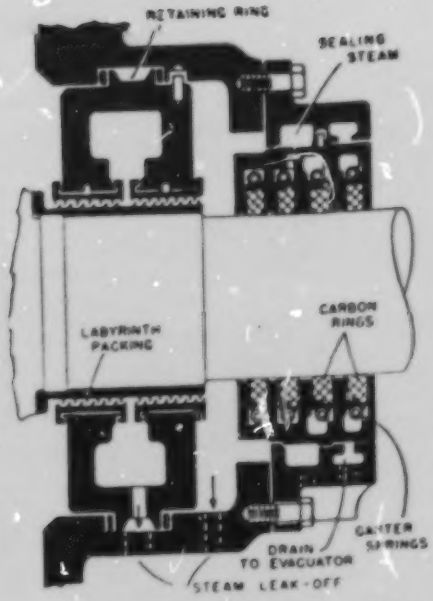



Low-Pressure
Labyrinth Shaft Seal
Fig. 25

In water seals a runner mounted on the turbine shaft acts in much the same way as the impellor on a centrifugal pump. A ring of water under pressure is maintained at the periphery of the runner, and this seals the gland against leakage. Clean water only should be used in a water seal since contaminated water will cause deposits which will build up and result in poor sealing and mechanical breakdown.

Turbine bearings in practically every instance are adjacent to the shaft seals. Therefore, if the seals are not in good condition the steam or water escaping along the shaft may find entry into the bearing pedestals and so contaminate the lubricating oil. Seals of all types are provided with adequate drains which should be operated as recommended by the turbine manufacturer to ensure that water will not accumulate in the cavities of the gland housing and leak into the bearings.

One or more steam "leak-off" points are usually incorporated in the seal design to enable steam leaking through the labyrinth to be led either to the L-p seals where it can be used as sealing steam or re-introduced to a L-p stage of the turbine so that the remaining potential energy can be converted into useful work.




Combination Labyrinth and Carbon Ring Shaft Seal
Fig. 26

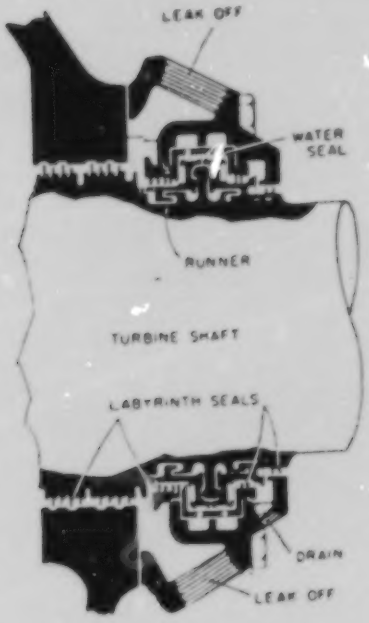
*The combination carbon ring* and labyrinth gland shown in Fig. 26 has two steam "leak-off" points for the removal of steam which has passed through the labyrinth part of the gland.

Referring to the figure, the labyrinth seal consists of two labyrinth rings each made of four segments and kept in place by a split retaining ring.

Fins on the bore of the labyrinth rings mesh between the fins formed on a sleeve on the shaft. Each of the four carbon rings is made of four segments held to the shaft by a circular garter ring.

Steam leak-offs to suitable stages in the turbine are provided between the two labyrinth packings and between the labyrinth and carbon ring seals.

L-p sealing steam is admitted between the second and third carbon rings.




Combination Water Seal and Labyrinth Seal
Fig. 27

Where water-type seals are provided the water should not be turned on until the turbine is running at above half-speed since below this speed the impellor will not create sufficient pressure to provide an efficient seal, see Fig. 27.

For the same reason water should be cut off from the seal as soon as the steam has been cut off from the turbine.

(PE2-3-4-24)

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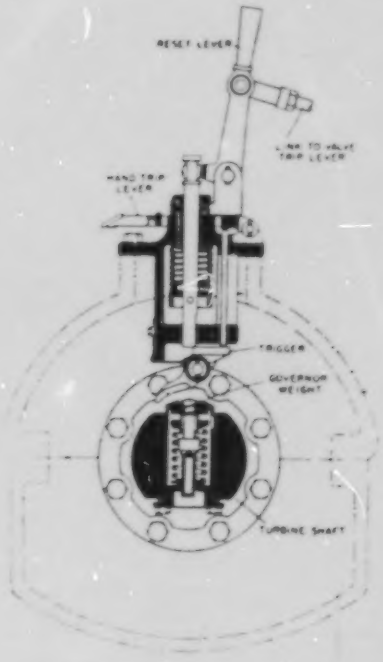
## Governors

The governing of all steam turbines is effected by controlling the amount of steam admitted to the turbine. This control is usually performed by two governors. One of these is designed to shut off the supply of steam in the event of the rotor speed increasing above a predetermined maximum. It is often referred to as an overspeed governor or emergency trip, Figs. 28 and 29.

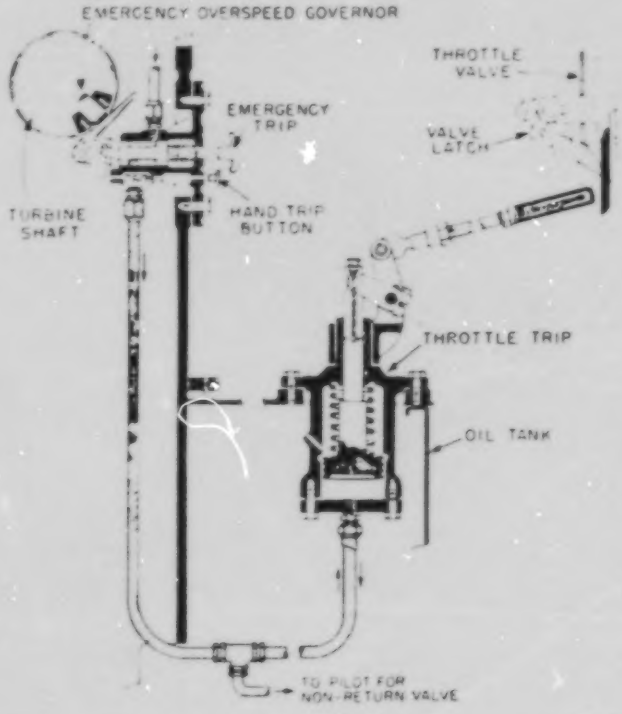
The other or main governor operates to maintain the turbine at a constant speed - or it may even be required to operate the turbine at varying speeds when acted upon by some outside influence.

Referring to Fig. 28, should the turbine speed exceed a predetermined figure, centrifugal force causes the governor weight to move out from its normal position in the turbine shaft and engage a trigger. This releases the spring-loaded linkage which operates a trip permitting the stop valve to close.

In the governor shown in Fig. 29, a latch holds a spring-loaded pilot valve so that H-p oil holds up a spring-loaded throttle trip piston. If the turbine speed exceeds the predetermined maximum, a plunger mounted in the rotor shaft is thrown out and trips the latch which holds the pilot valve. Upon release the valve closes off the supply of H-p oil and opens the line from the trip cylinder to the drain. This enables the spring to force the piston down and the mechanism holding the throttle valve open to unlatch.




Emergency Overspeed Governor
Fig. 28




Hydraulic Emergency Overspeed Governor
Fig. 29

(PE2-3-4-25)

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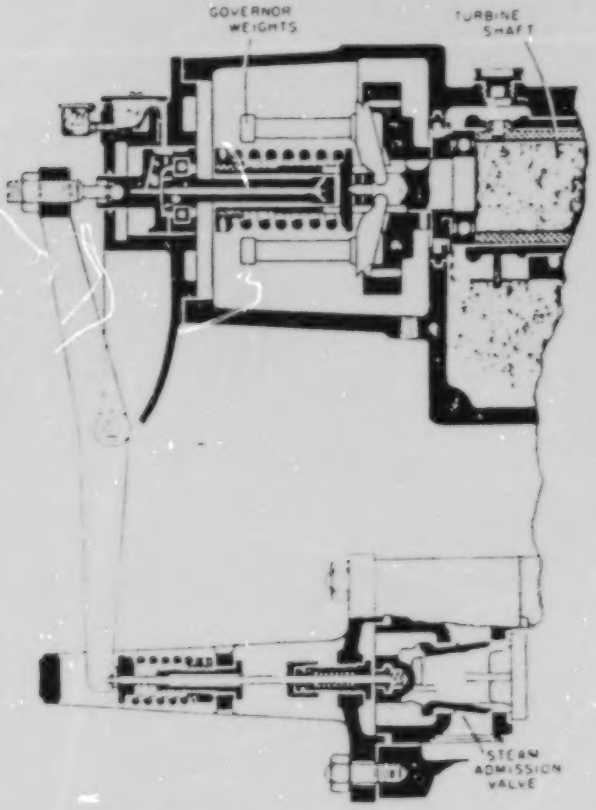
Extraction, mixed pressure and back pressure turbines are provided with governors which control the flow of steam in response to a combination of speed and one or more pressures. The governors of such units are exceedingly complex compared with the direct-acting speed governors such as one fitted to small mechanical drive turbines.

All governing mechanisms can be divided into three main portions - the governor speed-sensitive element; the linkage or mechanism which transmits the motion of the governor to the steam control valves, and the actual steam control valves. The mechanical details of governors will, of course, vary from maker to maker and only general principles can be described.

The most common type of speed-sensitive element is the more-generally used centrifugal unit or fly-weight governor. Weights pivoted on opposite sides of a spindle and revolving with it move outwards due to centrifugal force against the action of a spring as the turbine rotor speed increases, and inwards as the speed decreases.

This motion can be used to operate and control the steam admission valve directly through suitable link mechanisms, Fig. 30.




Mechanical Speed Governor
Fig. 30

Referring to Fig. 30, when speed increases, centrifugal force causes the governor weights, driven by the turbine rotor, to move outward against the spring pressure.

This movement, acting through a mechanical linkage, closes a balanced steam admission valve. This type of governor is only used on small type turbines.

The fly-weight governor will more often operate the pilot valve of a hydraulic system which will in turn admit and release oil under pressure to opposite sides of a power piston, or to one side of a spring-loaded piston.

Movement of this piston opens or closes the steam valve and thus control of the turbine speed is effected.

(PE2-3-4-26)

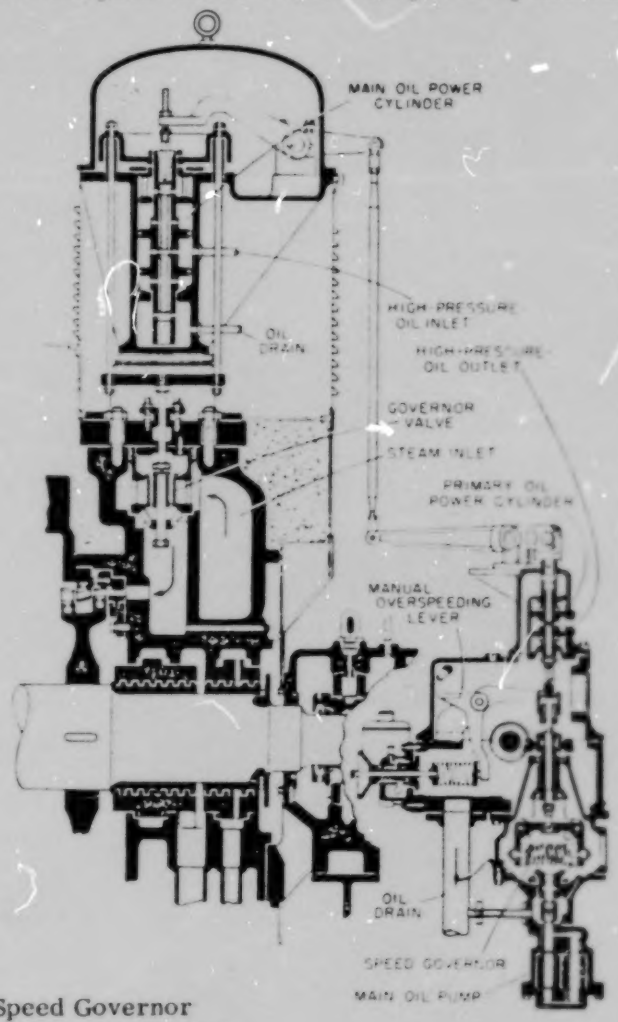
Some large high-speed turbines are provided with a double relay hydraulic system, such as shown in Fig. 31, the purpose being to increase further the force of the centrifugal governor, and also to decrease the time lag between the action of the governor and the change in rotor speed.

The centrifugal governor is worm-driven from the turbine rotor, so that when the turbine slows down centrifugal force acting on the fly-weights decreases and they are pulled inwards by a spring. This movement of the weights raises a pilot valve in the piston in the primary cylinder, which allows H-p oil to flow under the piston from the centre portion of the cylinder.

At the same time the space above the primary piston is connected to the drain. H-p oil under the piston causes it to move upwards and operate the link mechanism which is connected to the pilot valves of the main power cylinders (only one of which is shown). Upward movement of the primary piston causes upward movement of the main pilot valves, which admit H-p oil to the underside of the main power pistons and these in turn open the steam valves. The rocker arms and the pilot valve stems are so adjusted that the steam admission valves open and close in a predetermined sequence.

In small turbines the flow of steam is controlled by increasing or decreasing the opening of a single valve, usually of the balanced type, to reduce the force which the governor must exert.

In large units a number of valves are employed to regulate the steam flow, one for each group of steam nozzles. The number of valves and hence the number of nozzle groups in use are varied according to the load on the turbine.




Speed Governor with Double Relay Hydraulic System Fig. 31

(PE2-3-4-27)

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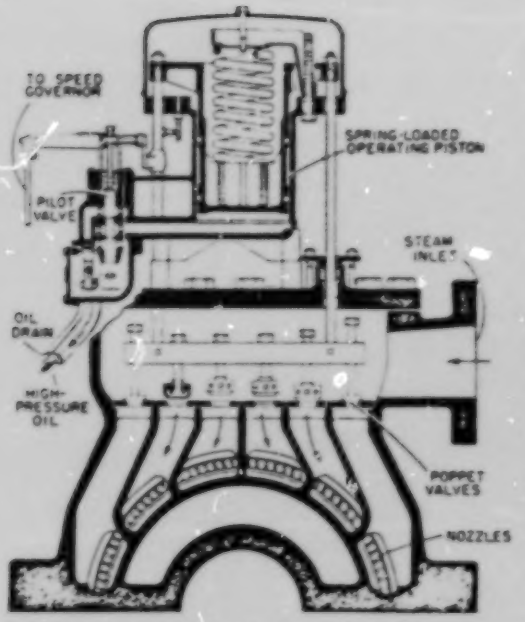
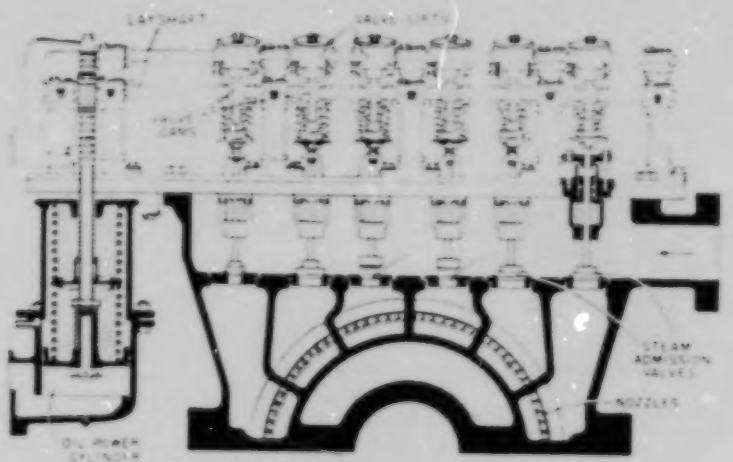



Bar-lift Steam Admission Valves
Fig. 32

These valves may be operated by a bar-lift mechanism, Fig. 32, by cams or levers, Fig. 33, or by individual hydraulic cylinders, already illustrated in Fig. 31. Sometimes hand-operated valves are provided to admit steam to additional nozzles or to a lower stage when greater than normal output must be developed.

In Fig. 32, when the speed governor calls for more steam, the pilot valve admits H-p oil to the underside of the spring-loaded operating piston. As the piston rises, it lifts a bar, which in turn lifts the poppet valves in a predetermined sequence. Each valve admits steam to one group of the first stage nozzles.

Fig. 33 illustrates how oil under governor control acts on the underside of the spring-loaded operating piston. As the piston rises, a rack on the piston rod causes a layshaft to rotate. On this layshaft are a number of cams - one for each admission poppet valve. Each cam operating through a follower and rocker arm actuates a steam valve which supplies a group of nozzles. The cams on the layshaft are indexed so that the valves are opened in a predetermined sequence and closed in the reverse order.



Cam-lift Steam Admission Valves
Fig. 33

(PE2-3-4-28)

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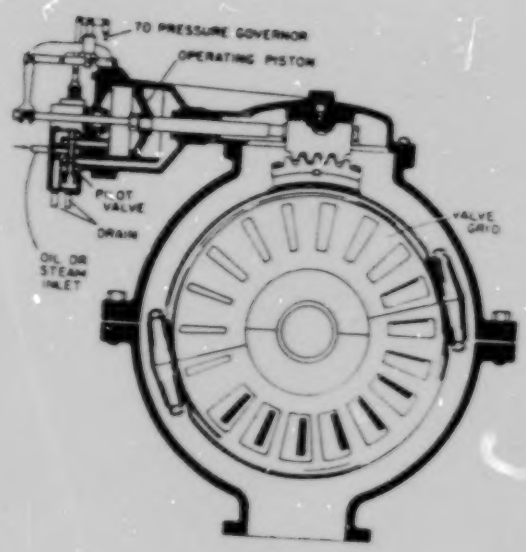
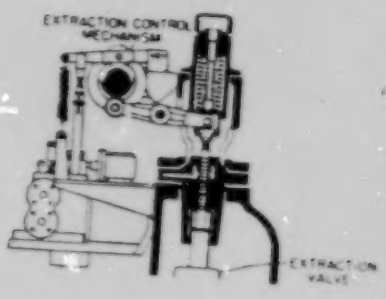
Extraction and back-pressure turbines are fitted with governors which are designed to maintain a constant extraction or exhaust pressure irrespective of the load on the turbine. The pressure sensitive element consists of a diaphragm or bellows and the response to pressure changes is communicated through linkages and a hydraulic system to the valves which control the steam extraction and also to the speed governor which controls the admission of steam to the turbine. On automatic extraction turbines the actions of the pressure and speed responsible elements are co-ordinated so that the turbine maintains the predetermined speed.

Extraction valves are designed to control the flow of steam to the stages following the point of extraction. Fig. 34 illustrates the grid-type extraction valve. It is placed within the turbine in the stage from which the steam is extracted and after the moving blades of that stage. It controls the flow of steam to the remainder of the turbine. Essentially, the valve consists of a ported stationary disc and a ported grid which may be rotated. When the openings in the disc and the grid coincide the valve is open and a full flow of steam passes [illegible] turbine. When




Grid-type Extraction Valve
Fig. 34

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## Speed Reduction Gear Sets

There are many instances where the most efficient and practical turbine speed is in excess of that of the machine being driven. Some of the more common examples of this include turbine-driven direct-current generators, paper-making machines, centrifugal pumps, blowers and fans. Under these circumstances reduction gear sets are used to reduce the speed of the turbine to suit that of the machine being driven.

Reduction gear sets used on modern medium and large-sized steam turbines are generally housed in an oil-tight casing and are connected to the turbine and driven unit by flexible type couplings. Small turbines may be designed so that the gear-set housing is integral with the turbine casing and the pinion may even be keyed directly onto the rotor shaft.

## Flexible Couplings

In some turbine generator sets the rotors are connected together by a solid coupling but it is more usual for the turbine to drive through a flexible or claw type coupling, Figs. 36 and 37. In most mechanical drive turbines and where a reduction gear set is involved the turbine is connected to the driven unit by a flexible type of coupling.

Flexible couplings permit an axial movement of the driven shaft and they can if necessary also be designed to transmit end thrust from the driven unit to the turbine. They can accommodate, to a minor extent only, the misalignment which may result from temperature changes, settling of foundations or bearing wear. They are not intended to overcome shaft misalignment due to careless or inexperienced erection.

The flexible couplings used on large direct-connected units, are enclosed in the same housing as the turbine and driven unit bearings, where they are lubricated by an oil bath carried in its own oil-tight case.

## Turning Gears

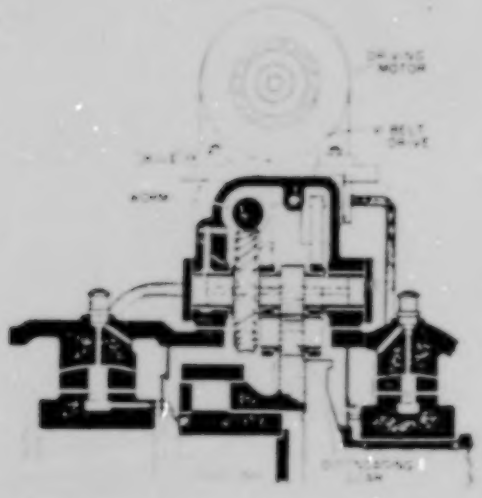
The distance between the bearings of large turbines is considerable and when operating temperatures are above 400°C, it becomes necessary to keep the shaft turning after shut down to make sure that uniform cooling takes place throughout the turbine. Uneven cooling may cause shaft distortion. A motor-driven turning or barring gear is often provided, therefore, to keep the rotor revolving at speeds up to 30 rev/min during the cooling period. The turning gear is also used when starting up.

It consists essentially of an electric motor connected by means of reduction gearing, to a gear ring either on the turbine shaft or coupling with a mechanism for disengagement when not in use, Fig. 38.

When the turbine revolves at slow speeds the main oil pump will, of course, not provide sufficient oil to lubricate the bearings. Consequently an auxiliary oil pump must be used when the turning gear is in operation. A separate motor-driven oil pump is generally provided to supply oil to the bearings instead of employing the turbine-driven auxiliary oil pump.

Some turbines are provided with a high-pressure motor-driven positive-displacement type pump, called the jacking pump, which supplies oil to each main bearing through a hole at the bottom of the bearing. The oil pressure lifts the journals on oil films before turning gear is brought into operation and this reduces the starting load on the electric motor. Once the shaft has started to rotate the high-pressure oil pump can be shut down.

In the turning gear illustrated below the motor speed is reduced by a belt-drive and a worm and wheel. The disengaging gear wheel is carried on a yoke which is slung from the worm shaft. An oil-operated piston is arranged to rotate the yoke about the worm shaft and so engage or disengage the turning gear from the turbine shaft.



Turning Gear
Fig. 38

(PE2-3-4-31)

## Steam Turbine Lubricating Systems

Turbines are the prime movers upon which the operation of a whole plant may depend and they must in consequence be provided with lubricating systems which will ensure a reliable supply of lubricating oil to all parts in motion. The size of the turbine is the main criterion in deciding whether the lubricating system shall be of simple or complex design. Small turbines of less than 150 kW such as are used to drive auxiliary equipment are normally provided with ring-oiled bearings, the other moving parts being lubricated by hand.

Moderate-sized turbines, particularly if driving through a reduction gear, may have both ring-oiled bearings and a circulating system which not only supplies oil in the form of a spray to the gears but also supplies oil to the bearings of the gear set and the turbine.

Large turbines are invariably provided with oil-circulating systems which supply oil not only to the turbine bearings but also to the governor mechanisms, the hydraulically-operated steam throttle valves and the bearings of the driven generators, etc.

## Thrust Bearings

In an impulse turbine the pressure of the steam drops in the stationary nozzles and therefore theoretically the steam pressures on both sides of the moving blades are equal. For this reason there is little tendency for the steam to exert an axial thrust on the shaft. However, there is always a small thrust in an impulse turbine which tends to displace the shaft in an axial direction. This thrust must of course be counteracted since failure to do so would result in contact between the moving and stationary parts of the turbine with disastrous results.

The reaction turbine presents an entirely different picture. In this instance there is a considerable pressure drop across each row of moving blades and as a result an end thrust is imparted to the turbine shaft by each row of blades. This thrust is in addition to the thrust which is developed by the rotation of the shaft. One method of reducing the end thrust to zero is the double-flow principle of turbine design in which steam is admitted to a point midway along the turbine casing, after which it divides and flows axially in both directions. Opposing rows of blading are mounted on either side of the steam inlet and hence the equal end thrusts developed in these blades counteract each other since they are in opposing directions.

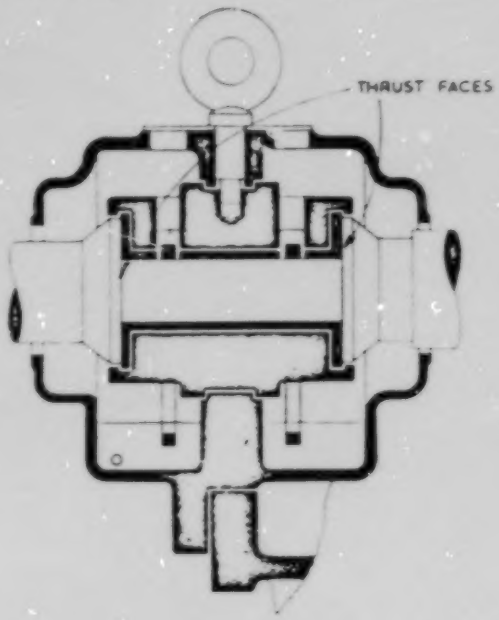
Another method of counteracting end thrust is by means of a balance piston upon which steam impinges and in doing so exerts a force equal and opposite in direction to the end thrust in the shaft.

Yet another system to eliminate end thrust in large compound reaction installations consists of coupling together in tandem the high and intermediate pressure units and causing steam to flow through each in opposite directions. The opposing thrusts developed balance each other and so render unnecessary the employment of large thrust bearings.

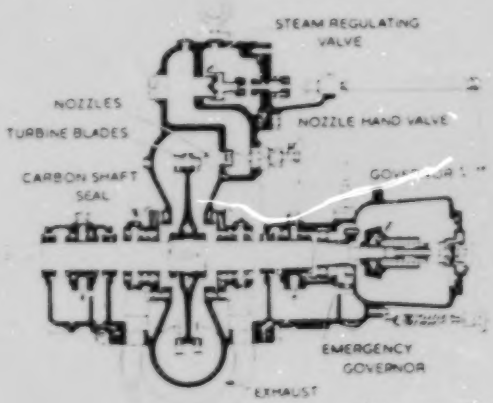
Regardless of the type of turbine and the degree of axial thrust, some type of thrust bearing is always provided on the shaft to maintain the correct axial position of the moving parts with respect to the stationary parts. Normally thrust resulting from the steam flow is towards the L-p end of the unit, but thrust bearings are always incorporated to prevent axial movement in both directions.

In small turbines the thrust and radial bearings are often combined in one single design as in Fig. 39. The bearing metal is extended radially over the ends of the bearing shell to form thrust bearing surfaces which are sometimes provided with oil grooves to permit a more efficient distribution of the lubricating oil.

Fig. 40 shows a ball bearing in a small mechanical drive turbine. The axial load component of the ball bearing is being employed in this case to maintain the rotor in its correct position with respect to the steam entry nozzles and the stationary blades.




Ring Oiled Radial Bearing with Simple Thrust Pads on its ends
Fig. 39

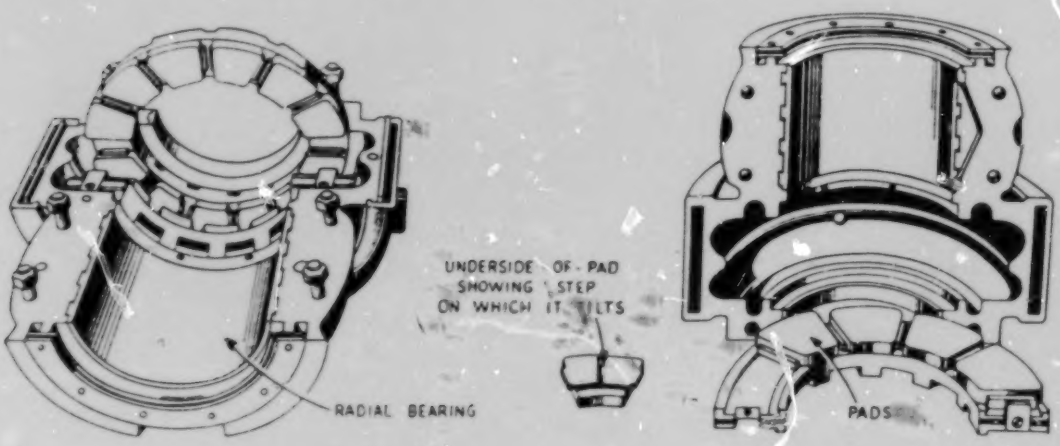



Mechanical Drive Turbine Fig. 40

(PE2-3-4-33)

Small units such as this are employed to drive pumps, fans and similar auxiliaries, often through reduction gearing. The turbine (Fig. 40) is a single-stage impulse type turbine with velocity compounding. It has a direct-action fly-ball governor and ring-oiled bearings.

In the Michell type of bearing, Fig. 41, pivoted pads adjust themselves to the wedging action of the oil between the bearing surfaces. The lubricant is drawn into the wedge-shaped space so formed and the H-p oil films generated between the surfaces eliminate all metallic contact and enable the thrust to be floated entirely on oil.




Combined Radial and Michell
Thrust Bearing
Fig. 41

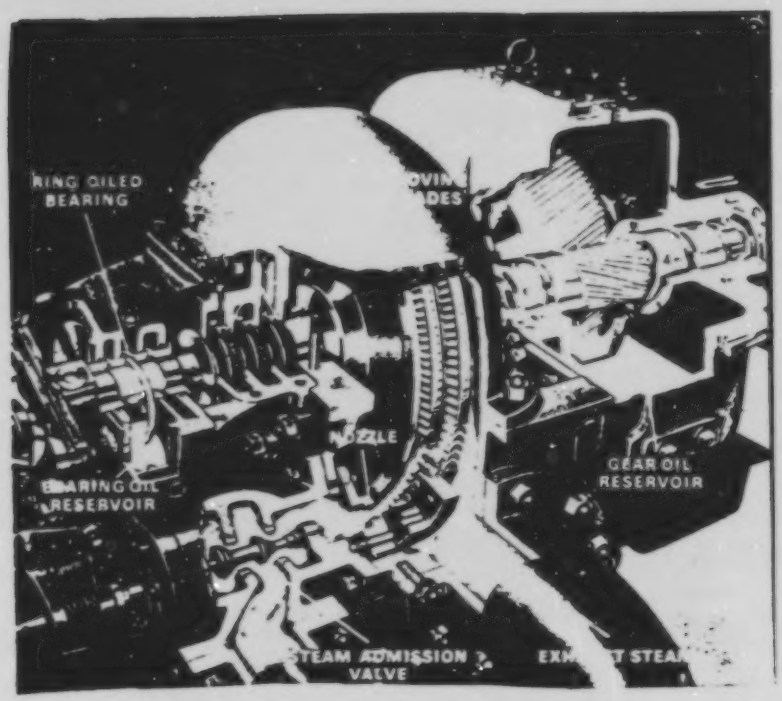
Ring-oiled Bearings

A cut-away section of a small turbine equipped with ring-oiled bearings is shown in Fig. 42. When the turbine is in operation the rings, which ride freely on the journals and revolve with them, dip into oil contained in the reservoir in the bearing housing. They automatically carry oil to the top of the journal from the reservoir, and it is then distributed over the whole length of the journal by rotation and by grooves in the bearing metal.

In the ring-oiled bearing shown in Fig. 43 axial grooves in the bearing metal are provided both ahead of and after the pressure area. This ensures complete oil distribution over the whole length of the bearing surface. Annular grooves are often provided close to the ends of the bearing to collect oil as it is squeezed out of the ends of the bearing. Through holes at the bottom of these grooves the leak-off oil is permitted to drain back into the bearing reservoir, thus reducing the considerable oil loss from the housing and the tendency to formation of oil mists.

The oil reservoir should be either fitted with a sight oil level gage or with a constant level oiler.

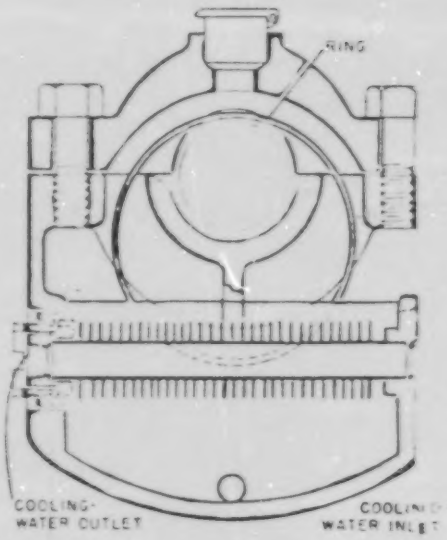
(PE2-3-4-34)




Cutaway Section of a small Turbine
Fig. 42

When the turbine is operating with high temperature steam, or is placed in an unusually warm location, the oil in the bearing reservoirs may become quite hot, and in order to maintain the oil at a steady operating temperature, cooling water jackets or coils are sometimes incorporated in the bearing design.

Referring again to Fig. 43, most small mechanical drive turbines are fitted with ring-oiled bearings. One or more rings rest on the journal and dip into the oil reservoir in the bearing base. Rotation of the journal also rotates the rings which carry oil from the reservoir to the top of the journal from where it is distributed to the bearing surface. In this particular design, provision is made for cooling the oil by water.




Ring-Oiled Bearing
Fig. 43

(PE2-3-4-35)

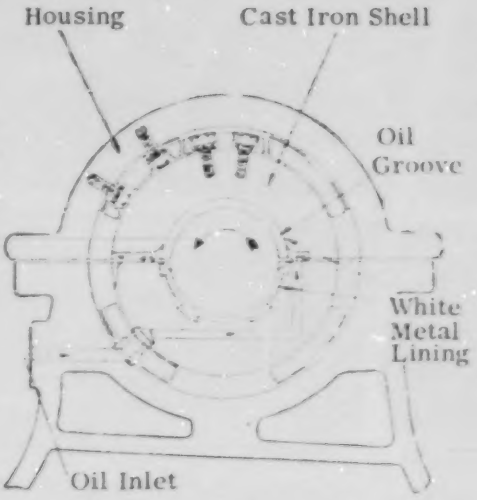
## Pressure-fed Bearings

The rotor of a steam turbine is supported by two main bearings both of which are outside the steam cylinder. Because of the extremely small clearances between the shaft and the shaft seals and between the blading and the stationary parts, the bearings must be accurately aligned. Wear must be at an absolute minimum for the same reason or damage will result to the shaft seals and blading.

The loads imposed upon the main bearings are chiefly due to the weight of the rotor assembly. This may or may not be equally divided between the bearings depending upon the relative position of the bearings and the center of gravity of the rotor assembly. However, the design is usually such that the bearings do take equal shares of the load. In turbines where the admission steam is not uniformly distributed around the circumference, the forces on the blades have an influence on the bearing loads and pressures. If these unbalanced forces become considerable a vibratory load may be imposed upon the bearing in addition to that imposed by the rotor weight.

Large turbine main bearings generally consist of shells split horizontally and lined with an anti-friction bearing metal. The bearings are enclosed in a housing to which a generous supply of oil is pumped by the circulating pump. This oil is delivered to the bearing, and chamfers and oil grooves assist in its complete and even distribution along the length of the journal. When an oil of correct viscosity is used a wedge is formed between the journal and the bearing - the journal floats on this oil wedge and metal-to-metal contact between the journal and bearing cannot occur.

The passages and grooves in the bearings are proportioned to permit a considerably greater flow of oil than is required solely for lubrication. This additional oil flow is required to remove the frictional heat and the heat conducted to the bearing by the shaft from the hot parts of the turbine. The oil flow must be sufficient to cool the bearing, prevent hot spots due to induced heat, and maintain the oil and the bearing at a proper operating temperature. In fact, the major portion of the oil supplied to turbine main bearings serves more as a cooling agent than as a lubricant.




Main Bearing
Fig. 44

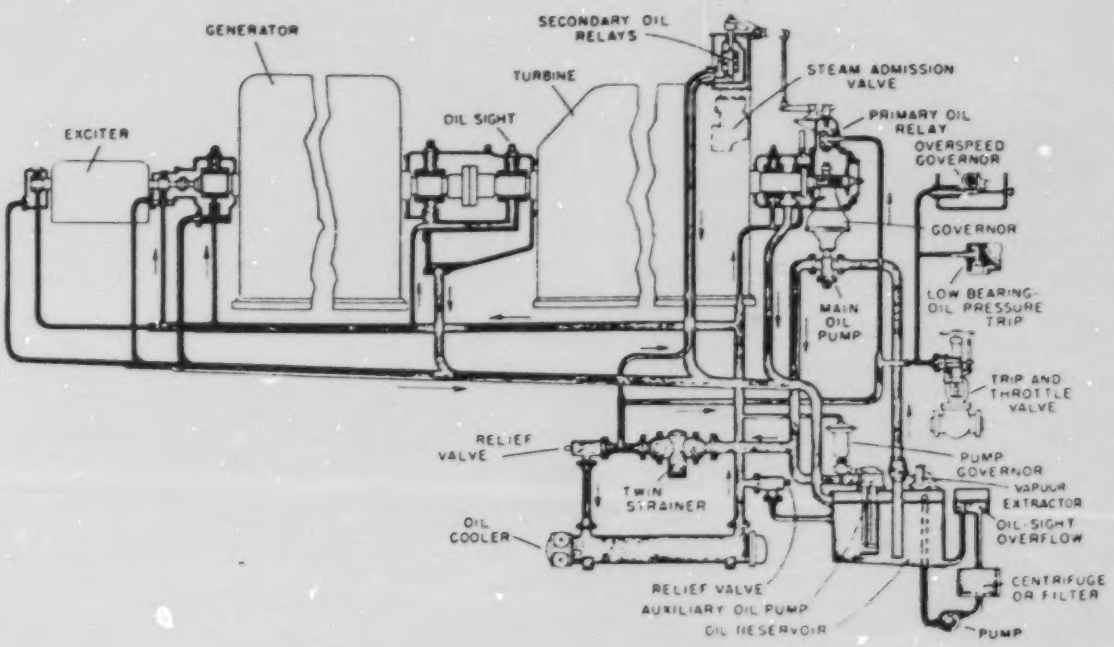
A thermometer is normally provided in each main bearing to allow the bearing temperature to be logged at regular intervals, a procedure which enables a very accurate check to be kept upon the condition of the bearing. A sudden rise in temperature will indicate a local condition needing attention.

Fig. 44 shows one type of main bearing used in modern steam turbines.

## Oil Circulating Systems

The details of oil circulating systems vary widely from turbine to turbine, the particulars depending upon the make, size and type of each individual unit. Their general arrangements are, however, similar, and the system shown in Fig. 45 may be regarded as typical.

Oil is drawn from a reservoir and delivered at full pump pressure 350 kPa to 500 kPa to a header which supplies oil to the governing and control mechanisms. Oil from this header, after being reduced in pressure to between 55 to 103 kPa, flows through an oil cooler to another header which supplies oil to all the bearings and other parts requiring lubrication. The oil returning from the bearings and governor mechanism drains back into the reservoir.




Turbine Lubricating Oil System
Fig. 45

(PE2-3-4-37)



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QUESTION SHEET

Second Class
Sect. 3, Lect. 1

POWER ENGINEERING

1. State briefly the essential difference between the principles of operation of impulse turbine blading and reaction turbine blading.

2. (a) What methods are used in impulse turbines to break down the steam pressure drop between turbine inlet and turbine exhaust, and why is this done?

   (b) It is common practice to use a velocity compounded impulse stage at the steam inlet or H-p end of a reaction turbine. Why is this done?

3. Give an example of force produced by reaction. Why is it not practicable to build axial flow industrial turbines to operate on this principle only?

4. What provision is made on a large turbine casing for expansion in longitudinal and radial directions? How should this be maintained?

5. List four types of turbine rotors. State in what types of turbines you would expect to find these in use.

6. (a) Why is a dummy piston essential in a reaction turbine but not in an impulse turbine?

   (b) Is there any end thrust in an impulse turbine due to steam flow? Explain.

   (c) Sketch a dummy piston, showing its position on a turbine rotor.

   (d) How is the position of the balance pipe in the turbine cylinder decided upon?

7. Turbine blading is basically of two types, namely impulse and reaction. These types are recognizable on sight. Sketch a section through each and name the type.

8. Is thrust-adjusting gear important in a reaction turbine?
   How would you set this equipment correctly?
   What effect would its incorrect setting or operation have upon the running of the turbine?

9. List the common methods in use today for sealing turbine glands. Sketch and describe each type.

(see over)

10. (a) What are the operating principles of speed-sensitive and pressure-sensitive turbine governors? When would each of these types be employed?

    (b) Describe the purpose and the operation of an emergency overspeed governor.

11. List three main advantages to be gained from fitting shaft-turning gear to a large steam turbine? What precaution must be taken before engaging the gear?

12. Sketch and describe a plain collar thrust and a Michel type thrust. Explain why the latter type can withstand a greater pressure.

INSTRUCTIONAL LEARNING SYSTEMS

# 8.3

## STEAM TURBINES -- AUXILIARIES

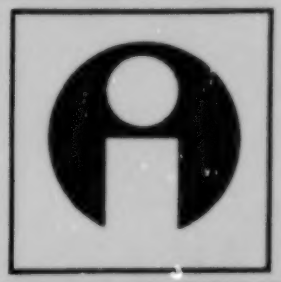
### Goal:

The apprentice will be able to describe steam turbine auxiliaries and their functions.

### Performance Indicators:

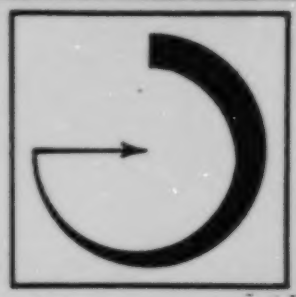
1. Describe condensers.
2. Describe feedwater heaters.
3. Describe deaerators.
4. Describe evaporators.
5. Describe cooling towers.

# Study Guide



* Read the goal and performance indicators to find what is to be learned from package.
* Read the vocabulary list to find new words that will be used in package.
* Read the introduction and information sheets.
* Complete the job sheet.
* Complete self-assessment.
* Complete post-assessment.

# Vocabulary



* Air ejector
* Air release valve
* Atmospheric relief valve
* Central or radial flow condenser
* Circulating water pumps
* Condenser gauge glass
* Condenser tubes
* Condensate pump
* Cooling tower
* Cooling water flow
* Deaerator
* Down flow condenser
* Electrical purity measurement
* Ejector condenser
* Evaporator
* Feedwater heater
* Forced draft cooling tower
* Hyperbolic draft cooling tower
* Induced draft cooling tower
* Jet condenser
* Mechanical draft cooling tower
* Non-return valve
* Regenerative condenser
* Relief valve
* Shell
* Silver nitrate test
* Surface condenser
* Tube plates
* Vacuum pay-off relays
* Vacuum trip relays
* Waterbox

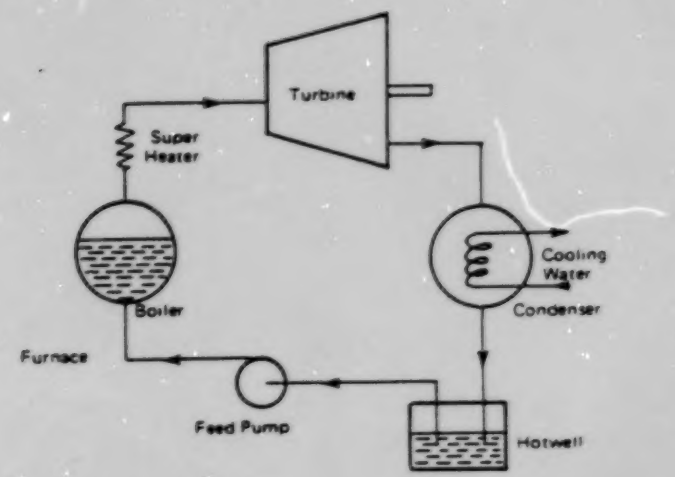
# Introduction

The steam turbine is the prime mover of a steam operated power plant. But the turbine must have the help of other equipment to complete its job of converting heat energy into mechanical energy. The equipment that helps convert steam into mechanical energy are called auxiliaries.

A steam plant operator must understand the operation of the turbine and the auxiliaries. This package is designed to acquaint the apprentice with steam turbine auxiliaries and their function in power production.

# Information

A simple steam plant is composed of the following components:




Steam from the boiler passes through a superheater into the turbine. The exhaust steam is transformed into water in the condenser and stored in the hotwell. A feed pump pulls water from the hotwell and supplies it as feedwater back to the boiler.

Condenser

The condenser is a heat exchanger. Its job is to convert exhaust steam to water so that it can be recompressed at boiler pressure. A surface condenser uses river water or a cooling tower to transform the exhaust steam into water. Another method of cooling involves air cooling of finned tubes that carry the steam. A condenser is made of the following parts:

- Shell of welded steel construction with attached hotwell, exhaust neck and support plates.
- Tube Plates made of brass or stainless steel.
- Condenser Tubes of small diameter brass or alloy which are attached or welded to the tube plates.

# Information

- Waterbox made of cast iron and bolted to the shell with tube-plate collar bolts.

- Cooling water flow that passes through the waterbox.

The efficiency of a condenser is affected by the arrangement of tube banks. If tubes are properly arranged, the condensate can be reheated and deaerated with steam in the condenser. A condenser that can utilize condenser steam for reheating the condensate is termed a regenerative condenser. A down flow condenser has a steam flow that is vertically downward. Central or radial flow condensers flows steam around the tube banks and radially to the center. Jet condensers use a water spray to cool the steam and both coolant and condensate flow into the hotwell. The ejector condenser is very much like a jet condenser. Exhaust steam enters the cooling water flow and is condensed by mixing.

Air ejectors are needed to remove air from the condenser. Air will build up and blanket the cooling surface. This reduces the efficiency of the condenser. The air ejector expands high pressure steam through a nozzle which converts the heat to kinetic energy. The air ejector jets trap air and remove it from the condenser.

A condenser has a number of safety fittings. The atmospheric relief valve releases pressure when pressures within the shell become greater than atmospheric pressure. This prevents rupture of the shell. Large condensers use vacuum pay-off relays and vacuum trip relays to protect against excess pressure in the condenser. A condenser gauge glass shows the level of condensate in the condenser hotwell. Excess water levels in the hotwell can be detected by a high water level alarm. Detection of leaks in the cooling water can be detected by an electrical purity measurement with a dionic tester. A silver nitrate test will detect salt in the cooling water. Manufacturer's instructions for specific condensers should be carefully followed in operating a condenser. Circulating water pumps for moving cooling water are usually a vertical, propeller type, or mixed flow pump. Centrifugal pumps are used with large condensers. Condensate pumps remove the condensate from the hotwell to the aerators. Most condensate pumps are centrifugal type.

Feedwater Heaters

In bleeder trubines, steam is drawn off for the purpose of heating feedwater. As a result of bleeding, less exhaust steam is delivered to the condenser and the efficiency is increased. Feedwater heaters are used to help capture the energy that is normally lost as steam meets the cooling water. There are two

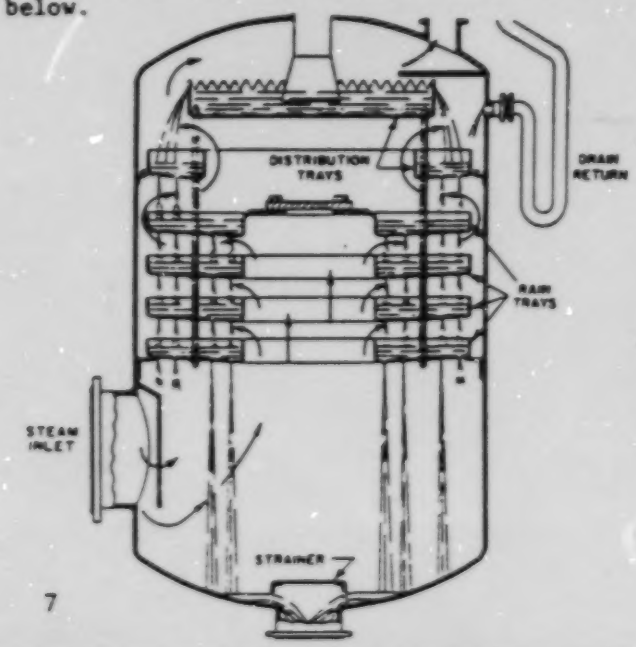
# Information

classes of feedwater heaters--low pressure and high pressure types. A feedwater heater has several safety and operational valves:

- Safety valve on the steam side to avoid overpressure problems.
- Relief valve on the waterbox to prevent excess pressure from thermal expansion of water.
- Non-return valve to prevent steam from returning to the turbine.
- Drain valve for draining off condensate on steam side. Another drain valve on waterbox for draining water.
- Air release valve on steam side to bleed off excess air that blocks entry of steam.

## Deaerators

A dearator removes the air from the condensate and heats it at the same time. Dearators are often called deaerator-heaters. The condensate is heated to the boiling point which releases all gases. After the condensate is heated, it flows down over a series of trays. The condensate flows to the bottom of the tank and the gases move to the top where they are vented off to the condenser. A tray type aerator is shown below.

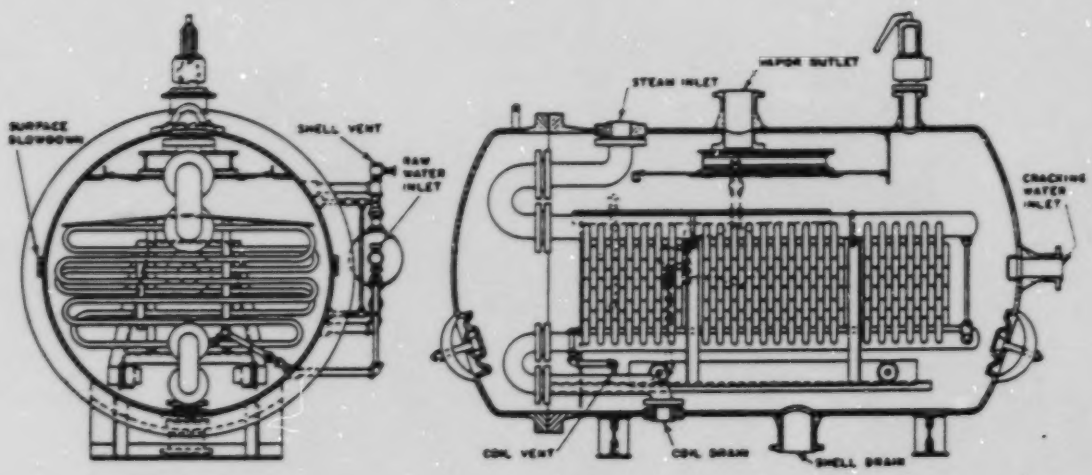



7

# Information

## Evaporators

Boilers require a pure feedwater. The feedwater must be free of minerals. The best supply of pure feedwater can be obtained from bleed steam evaporators. Bleed steam is directed at evaporator coils which produce a vapor. The vapor is condensed in a low pressure feedwater heater. The evaporator shell is made of steel and contains a coil and header assembly. The coil is looped inside the shell. Baffles on the coils separate water from vapor. As the water evaporates, the solids (mineral portion) of water are left in the evaporator. The evaporator must be cleaned regularly to remove scale and solids from the evaporation process. Clean surfaces offer better transfer of heat.




## Cooling Towers

In some settings, cooling water must be used over and over. This requires that the water by cooled after each use. A cooling tower or cooling pond is a common method for re-cooling water. In a cooling tower, the warm water is pumped to the top of the tower and allowed to drop over a series of splash bars. The water returns to the reservoir by gravity flow and is cooled along the way. A hyperbolic draft tower provides a chimney type suction that moves air past the cooling water.



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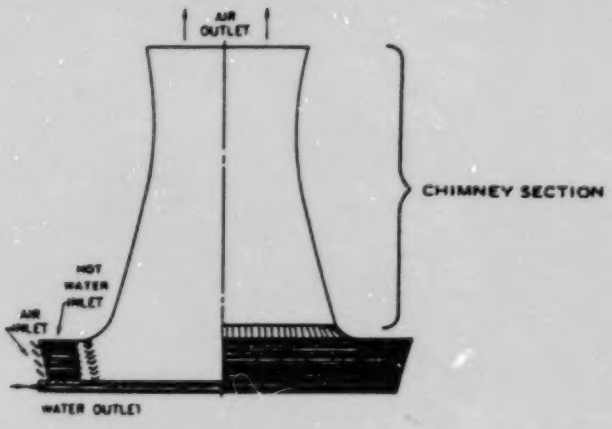
# Information




A mechanical draft tower forces air through the tower by a fan. If the fan is located at the base, it is a forced draft type that pushes air toward the top of the tower. An induced draft type has a fan located at the top of the tower and pulls air from the bottom.

# Assignment

* Read pages 1 - 37 in supplementary reference.
* Complete job sheet.
* Complete the self-assessment and check answers with answer sheet.
* Complete the post-assessment and ask instructor to check your answers.

# Job Sheet

ANALYZE A POWER PLANT FOR AUXILIARY EQUIPMENT

* Obtain permission to inspect a power plant.

* Inspect the auxiliaries that support the turbine and boiler.

* Identify by observation and interviews with employees:

  - Type and ports of the condenser
  - Air injector location
  - Safety fittings location and function
  - Feedwater heater arrangement
  - Deaerator arrangement
  - Evaporator arrangement
  - Use of cooling towers or ponds and their arrangement

* Sketch the flow of steam through the power plant boiler, turbine and auxiliaries.

# Self Assessment

Match the following terms and descriptive phrases.

| | |
|---|---|
| 1. Condenser shell | A. Used with bleeder turbines. |
| 2. Waterbox | B. Uses chimney type suction for air flow. |
| 3. Tube plates | C. Removes air and gases from condensate. |
| 4. Feedwater heaters | D. Has a fan located at bottom. |
| 5. Deaerator | E. Made of welded steel construction. |
| 6. Air ejector | F. Removes mineral from feedwater. |
| 7. Evaporator | G. Made of cast iron. |
| 8. Condenser gauge glass | H. Made of brass or stainless steel. |
| 9. Hyperbolic cooling tower | I. Shows level of condensate in hotwell. |
| 10. Induced draft tower | J. Removes air from condenser. |

# Self Assessment Answers

1. E
2. G
3. H
4. A
5. C
6. J
7. F
8. I
9. B
10. D

# Post Assessment

1. A _____ pump pulls water from the hotwell and supplies it as feedwater to the boiler.

2. A _____ condenser uses river water or a cooling tower to transform exhaust steam into water.

3. A _____ condenser utilizes condenser steam for reheating the condensate.

4. _____ condensers use a water spray to cool the steam.

5. Cooling water leaks can be detected by an _____ _____ measurement with a dionic tester.

6. Salt contamination of cooling water can be detected with a _____ _____ test.

7. Most condensate pumps are _____ type pumps.

8. A _____ removes air from the condensate and heats at the same time.

9. The best supply of pure feedwater can be obtained by using _____ _____ evaporators.

10. A mechanical draft cooling tower with a fan located in the base is a _____ draft type.

# Instructor Post Assessment Answers

1. Feed pump
2. Surface
3. Regenerative
4. Jet
5. Electrical purity
6. Silver nitrate
7. Centrifugal
8. Deaerator
9. Bleed steam
10. Forced

# Supplementary References

* Correspondence Course. Lecture 2, Section 4, Third Class.
Southern Alberta Institute of Technology. Calgary, Alberta, Canada.

SOUTHERN ALBERTA INSTITUTE OF TECHNOLOGY
CALGARY, ALBERTA

Power Engineering Department
Correspondence Courses

SECTION 4

PRIME MOVERS and AUXILIARIES

Third Class
Lecture 2

## STEAM TURBINE AUXILIARIES

### INTRODUCTION

A very large proportion of the electrical energy generated throughout the world is still produced through the medium of steam in power stations, in which the generators are steam driven. There are many plants using water turbines, internal combustion engines or gas turbines, but the steam turbine still remains the most important prime mover of all. The recent development of nuclear power stations has provided an entirely new means of heat supply but to date this heat supply has been utilized to produce steam for use in steam turbines.

Steam turbines have been described in an earlier lecture, this lecture will discuss some of the items of plant which serve as auxiliaries to these prime movers.

Fig. 1 shows a simple steam power plant in diagrammatic form. Heat released in the boiler furnace, is transformed through the medium of steam produced in the boiler, into mechanical work at the turbine shaft.

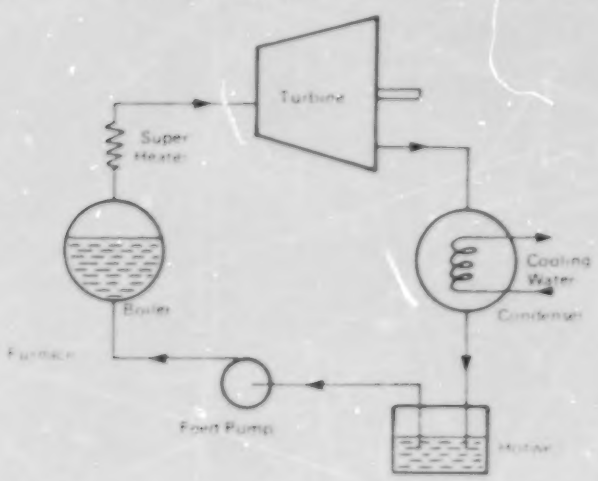



Fig. 1

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Fig. 2 shows a similar cycle but using a nuclear reactor as the heat source. The working fluid in the cycle is steam flowing through the turbine in the same manner as in Fig. 1. The heat exchanger supplies heat to the steam in place of the steam plant boiler.

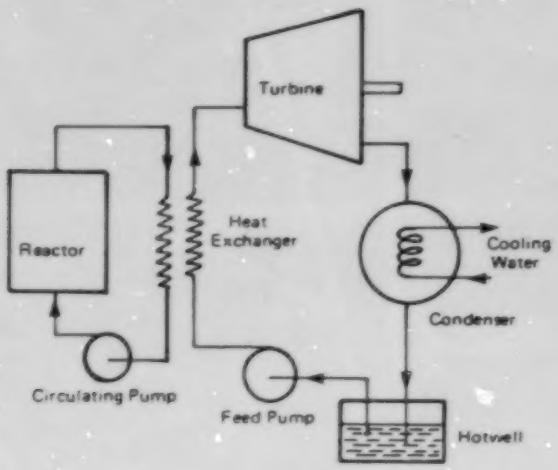



Fig. 2

The heat source for the cycle is derived from the "burning" of nuclear fuel in the reactor and this heat is transferred to the heat exchanger by a circulating coolant.

The prime purpose of the steam in any power plant is to supply the turbine with heat energy which it can transform into mechanical power. However, there are several auxiliary items depending upon steam supply, which are essential to efficient and economic turbine operation.

Examples are feedwater heaters, evaporators and deaerators. Other equipment operating on a similar principle includes condensers, air heaters, oil coolers, etc.

A convenient group title for this type of auxiliary plant item would be a "heat exchanger". The following will apply generally to heat exchangers; the individual plant items will then be described in turn.

In a power plant it is often necessary to warm up or cool down one substance or another; this involves the transfer of heat from one fluid to another in a heat exchanger. The word fluid in this sense means water, steam, gas, air - it is not confined to liquid.

( PE3-4-2-2 )

In almost all cases these heat exchangers operate on a surface heat transfer principle. That is, the two fluids are separated by a conducting material such as brass, copper or cupro nickel, and do not come in direct contact with each other.

The basic principle upon which all heat exchangers work is as follows:

1. There must be a difference in temperatures between the hot and the cold fluid.
2. The heat transfer will always take place from the hotter to the colder body.
3. The rate of heat transfer will depend on the temperature difference, the thermal resistance to heat flow of the material separating the two fluids, and the area of transmitting surface considered.

Expressed as an equation this can be written:

$$Q = k \times A \times \text{Temperature difference}$$

where Q is heat transfer, kJ/unit time

A is area considered in square metres

k is the thermal conductivity of the material

What does this mean in practical terms?

Consider the case of a condenser which is receiving the exhaust steam from a turbine.

Q ( heat transfer ) is the heat being transferred from steam to cooling water.

A is the surface area of the condenser tubes through which the heat passes.

k is the thermal conductivity of the condenser tube walls ( together with any films of scale, etc ).

Finally, the temperature difference is that between incoming exhaust steam flowing over the condenser tubes and the cooling water flowing through the condenser tubes.

( PE3-4-2-3 )

If the load on the turbine increases, the quantity of steam flowing and consequently the heat to be transferred, Q, increases.

In order to balance this increase there must be an equivalent increase on the other side of the equation. The thermal conductivity of the condenser tubes cannot increase nor can the cooling area, so the temperature difference from steam to water must increase.

The available cooling water inlet temperature however, will be unable to decrease so that in fact the necessary increase in temperature difference appears as an increase in the turbine exhaust temperature.

This is the same as saying that the turbine exhaust pressure has increased or that the condenser now operates at a reduced vacuum.

In most plants the following items of plant auxiliaries are to be found; the subsequent paragraphs will describe each in turn.

THE CONDENSER

The largest heat exchanger in the plant is the condenser. It is necessary for practical reasons, to condense the steam to water after completion of its work in the turbine, because only in liquid form ( water ) can it be recompressed to boiler pressure.

This necessary condensation of the steam causes the biggest single heat loss in the steam power cycle. The latent heat contained in the steam entering the condenser is transferred to the cooling water and dissipated to the atmosphere via the cooling tower or river.

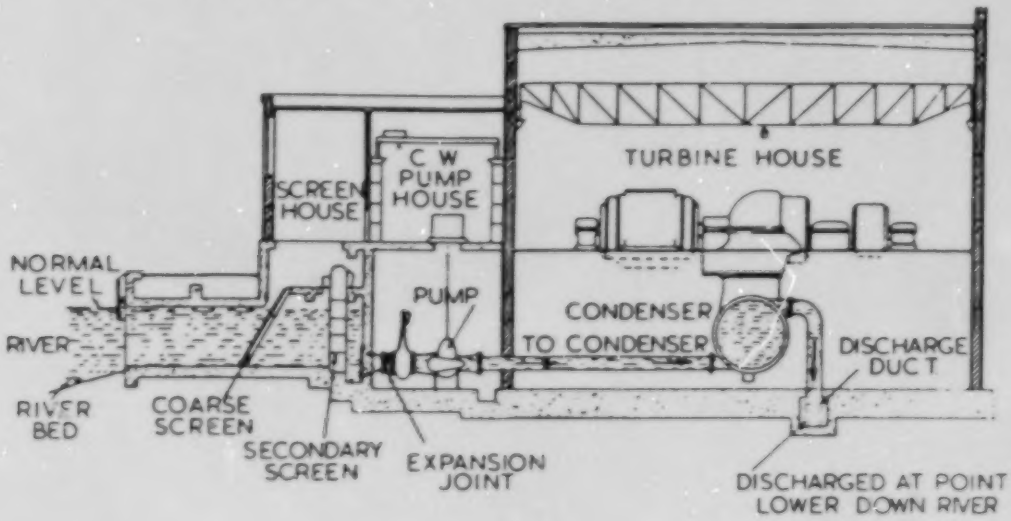
The above applies to a plant working on a condensing cycle. If the exhaust steam from the engine or turbine could be usefully employed for heating or process work, the overall plant thermal efficiency would become considerably higher. Typical figures for these plant thermal efficiencies might be 20% when exhausting to the atmosphere, 30% when condensing and 80% or even higher when the exhaust heat is used for heating or process.

There are several types of condensers in use, the most common being the surface condenser.

## The Surface Condenser

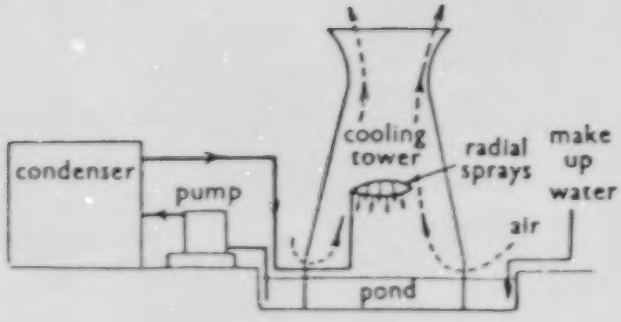
The main purposes of a surface condenser can be stated as:

1. To produce and maintain a vacuum at the turbine exhaust, thus allowing the steam to expand down to a lower pressure and so do more work.
2. To conserve pure boiler feedwater by condensing the exhaust steam which is subsequently returned to the boiler.
3. To act as a deaerator by removing air and other gases from the condensed exhaust steam.




Condenser using Cooling Water from River

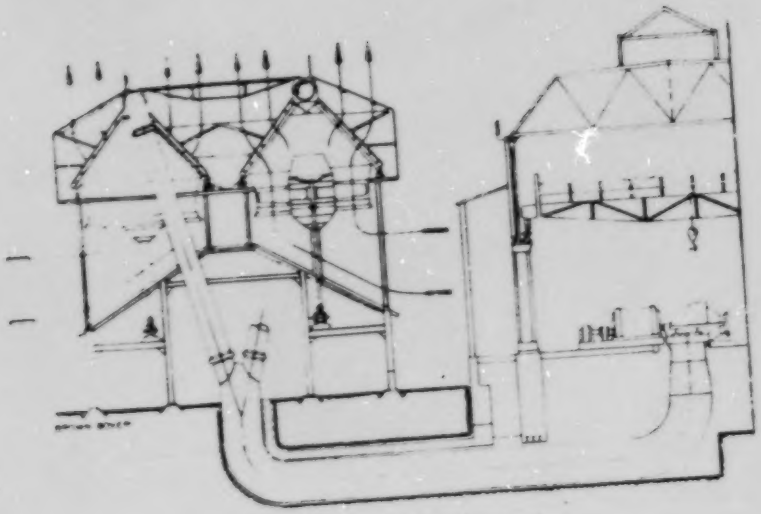
Fig. 3




Condenser using Cooling Water from Cooling Tower

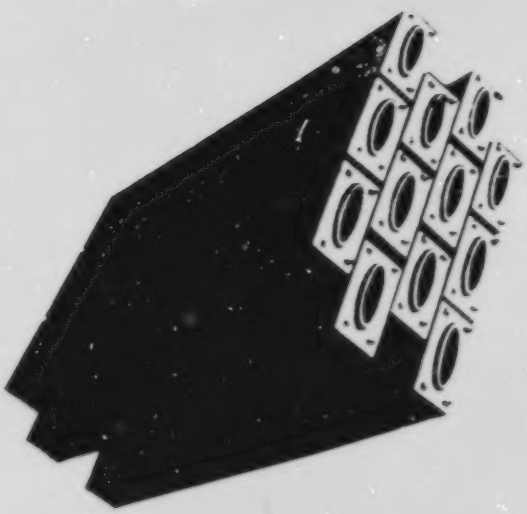
Fig. 4

( PE3-4-2-5 )



Air Cooled Condenser

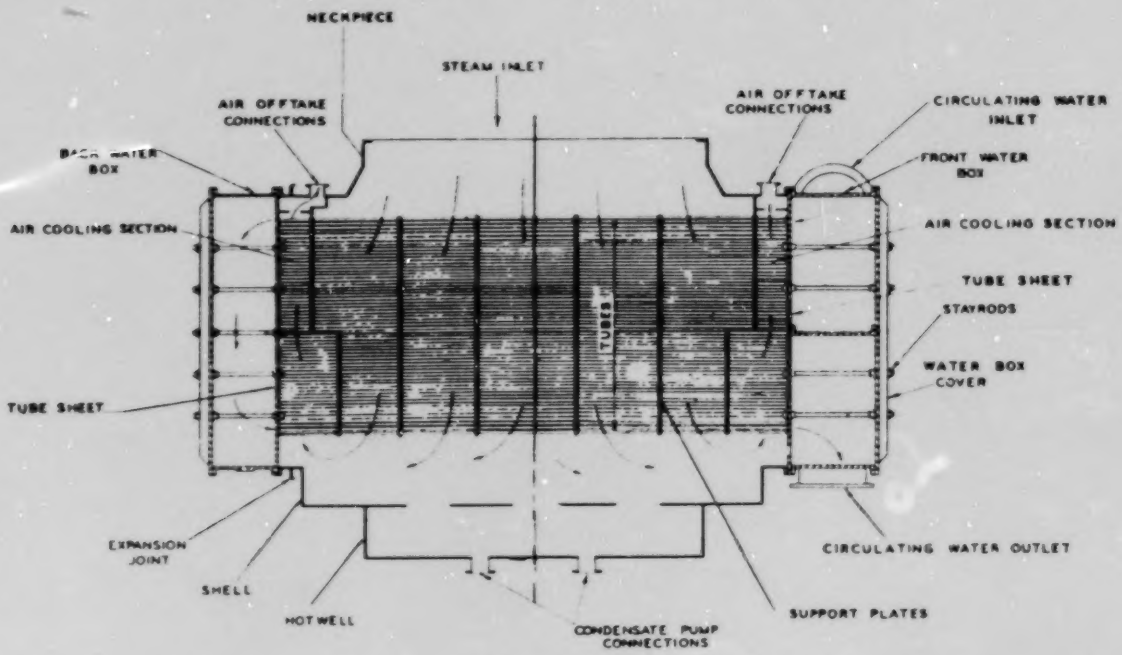
Fig. 5



Nest of Finned Tubes

Fig. 6

Fig. 3 shows a plant with a surface condenser using river water for cooling. Fig. 4 shows the layout of a condensing plant using circulating water from a cooling tower. In Fig. 5 the condenser is air cooled using finned tubes shown in Fig. 6; the steam flows inside the tubes and air flows around the tubes.




Surface Condenser

Fig. 7

Fig. 7 shows one form of condenser design and names the various parts.

1. The Shell

The shape of the shell may be cylindrical, oval or for large shells, rectangular. The shells are of welded steel construction reinforced with external ribs. The exhaust neck and hot well are welded to the shell, supporting plates welded to the shell support the shell as well as the tubes and dampen the tube vibrations.

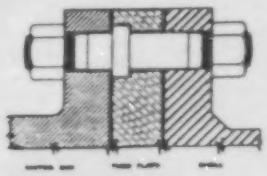
The condenser feet welded to the shell are, for small condensers, bolted solidly to the foundation and expansion bellows between turbine and condenser allow for thermal expansion and contraction.

(PE3-4-2-7)

The best solution for large condensers is to mount the condenser on flexible mountings and couple it rigidly to the turbine. In the case of a large turbine where a flexible coupling is used, the forces due to vacuum are often considerably greater than the weight of the condenser and therefore a rigid coupling is preferable.

2. Tube-plates

Condenser tube-plates are usually Admiralty brass ( 71% Cu + 28% Zn + 1% Sn ) or Muntz metal ( 60% Cu + 40% Zn ) 25 to 40 mm thick. They are bolted to the shell flanges with collar bolts as shown in Fig. 8, allowing removal of the waterboxes without disturbing the joint between shell and tube-plate.



Detail of Collar Bolt

Fig. 8

The tube-plates of brass or Muntz metal for large condensers are very expensive and may not be available in the sizes required. This problem has been solved with the application of welded steel plates with stainless steel cladding ( Fig. 9 ).

*Water-box of welded fabrication showing the tube-plate welded in position and the protective coating on the water side*

The tube-plate is clad with stainless steel on the water side.

1 = Water-box
2 = Tube-plate
3 = Stainless steel cladding
4 = Protective coating

*A* = Water side
*B* = Steam side

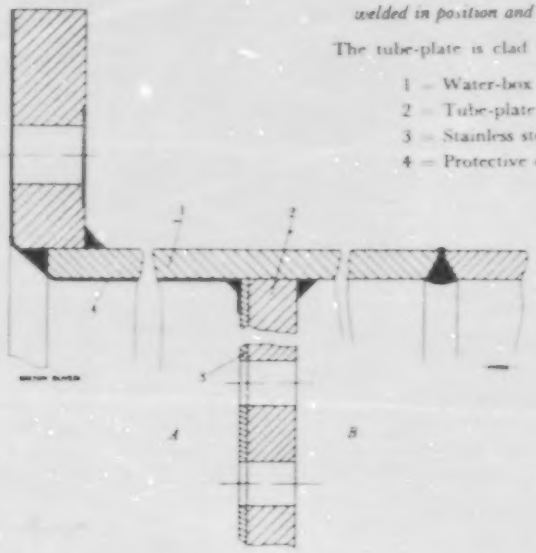
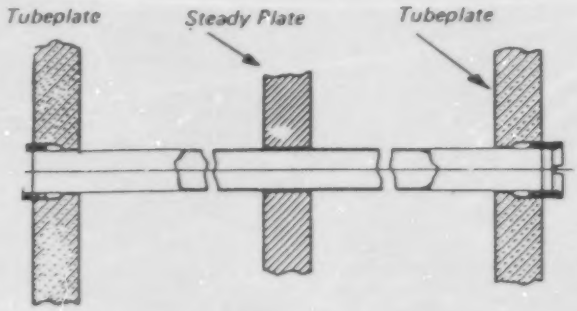


Fig. 9

( PE3-4-2-8 )

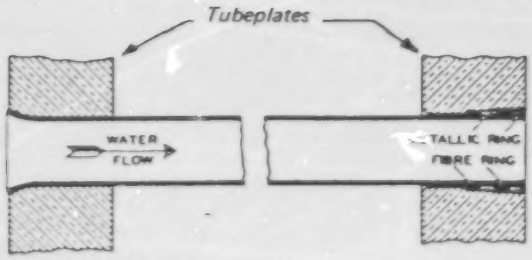
3. Condenser Tubes

A large heating surface can best be obtained with small diameter tubes. These also give the best heat transfer as the required wall thickness is very small. The tube material must be corrosion resistant and must be a good conductor of heat. The tube material may be Admiralty brass, aluminum brass ( 77.5% Cu + 20.5% Zn + 2% Al ) or cupro-nickel ( 70 - 90% Cu + 30 - 10% Ni ). Condenser tubes are available in sizes of 16 - 19 - 22 and 25 mm. The larger the condenser the larger would be the tubes used.




Tube secured at both ends with Ferrules

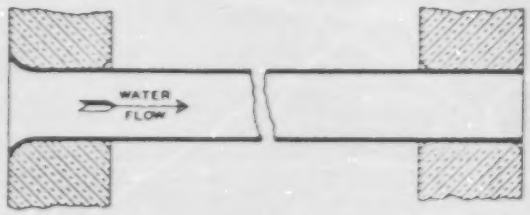
Fig. 10




Tube expanded one end only

Fig. 11

The tubes can be installed with ferrules, metallic or fiber packings, by roll-expanding, or with combinations such as inlet end expanded and bolled, outlet end packed or ferruled, ( see Figs. 10, 11 and 12 ) In some condensers the tubes are welded into the tube plates when tubes and plates are of practically the same material such as Admiralty brass tubes

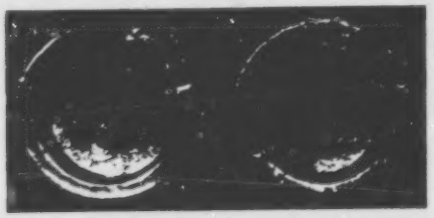



Tube expanded both ends

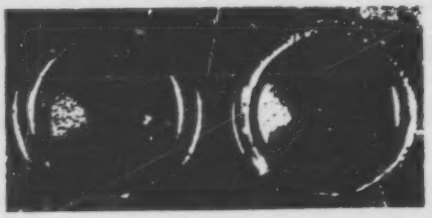
Fig. 12

( PE3-4-2-9 )

in Admiralty brass plates or cupro-nickel tubes in cupro-nickel plates. See Fig. 13.



Admiralty Brass Tubes in Naval Brass Tube Sheet.



Cupro Nickel Tubes in Cupro Nickel Tube Sheet.

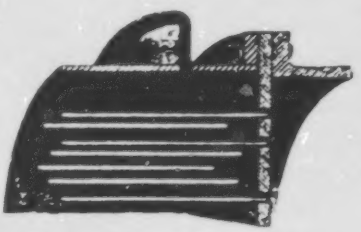
Appearance of Typical Welds

Fig. 13

Allowance must be made for differential expansion of tubes and shell. Packed tube-ends may allow the tube to move axially in the packing. For expanded or welded tubes the shell is usually equipped with an expansion joint ( Figs. 14 and 15 ).

Detail of Condenser Shell Expansion Joint

Fig. 14



Shell Expansion Joint

Fig. 15

4. Waterboxes

The classic material for the waterboxes has always been cast iron. Waterboxes are usually bolted to the shell with the tube-plate collar bolts ( Fig. 8 ). Hinged end covers allow for tube replacement, inspection, tube-cleaning and tube-plugging.

The waterboxes for large condensers are, for economical and practical reasons, of fabricated steel, and welded to the shell or they may be an integral part of the shell as shown in Fig. 9 where the waterbox has a coating of rubber or glass-fibre reinforced epoxy resin as corrosion protection, because in this case seawater is used as cooling water.

Stay-rods between tube-plate and waterbox end cover support both against the forces of the cooling water pressure ( Fig. 7 ).

5. Cooling Water ( C.W. ) Flow

The condenser may be designed for counter flow where the C.W. enters at bottom and discharges at the top of the waterbox, while the steam flows down through the condenser. Another design features parallel flow with C.W. entering the top and discharging at the bottom of the waterbox and this method reduces sub-cooling of the condensate ( Fig. 7 ). Double pass C.W. is the most common for small and medium size condensers, as this design gives a fairly short condenser of large diameter ( Fig. 7 ). When the condenser tubes exceed 7 - 8 metres it is more practical to make single pass, thereby reducing the height of large condensers.

Most medium and large condensers have double flow C.W. with vertical divided waterboxes making it possible to shut down one half of the condenser, while operating on the other half at reduced load. This allows tube cleaning without shut downs.

6. Arrangement of Tube Banks

The performance of the condenser depends to a large extent on the arrangements of the tube banks.

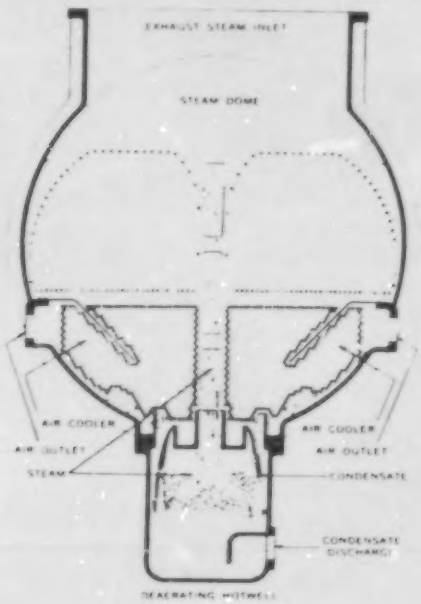
Air pockets between tubes must be avoided as the air prevents steam from coming in contact with the tubes and thereby taking heating surface out of service; the same goes for tubes submerged in condensate.

Sub-cooling of condensate must be avoided for the sake of heat losses and poor deaerating as sub-cooled condensate quickly absorbs oxygen.

The condensate around the tubes will always be sub-cooled, or all further heat transfer would cease due to lack of difference in temperature between steam and condensate. Proper tube arrangement makes it possible to reheat and deaerate the condensate with available steam in the condenser, and this type of condenser is termed " regenerative ".

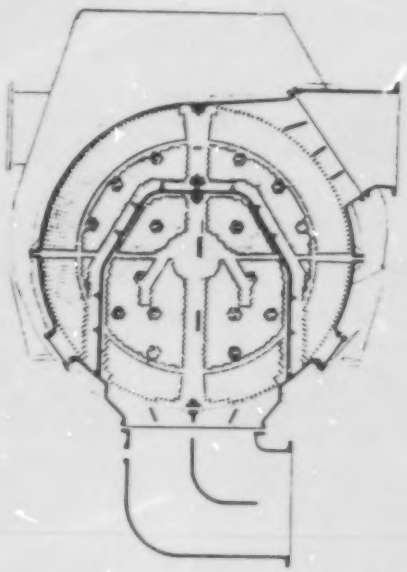
7. Down Flow Condensers

Fig. 16 shows a down flow condenser where the steam flows vertically down, steam lanes are provided to allow steam to flow to near the bottom of the condenser for reheating of condensate. The heavy concentration of tubes under the air suction baffles prevents steam from entering the air ejectors.




Down Flow Condenser

Fig. 16



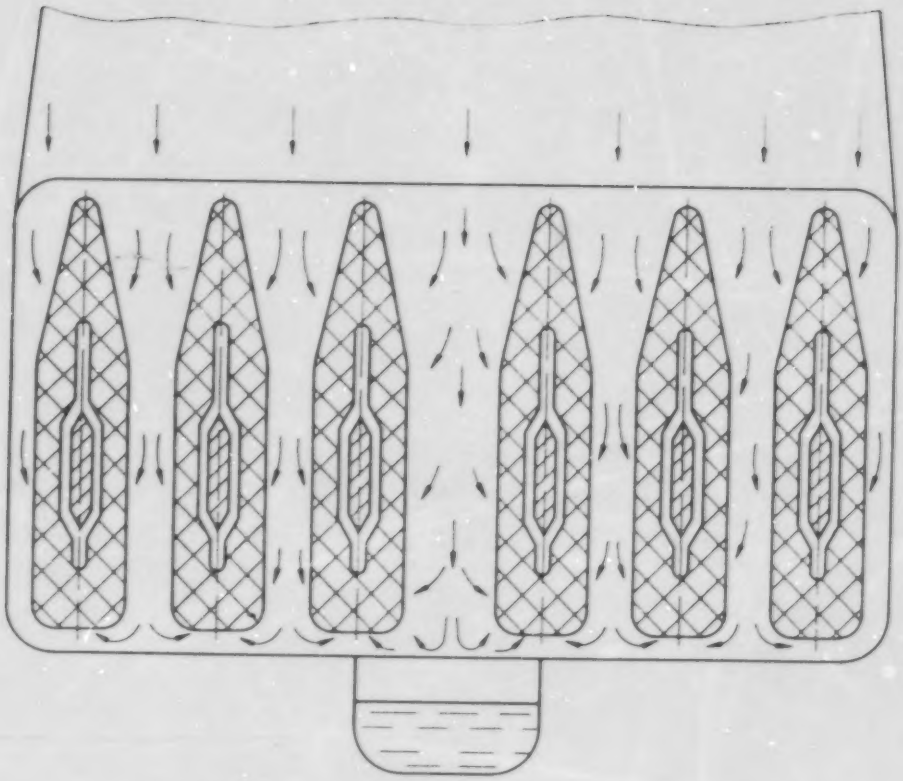
Central Flow Condenser

Fig. 17

8. Central or Radial Flow Condensers

A central or radial flow condenser is shown in Fig. 17. The steam flows around the tube bank and radially in against the centre and, as air always follows the steam, the air suction is from the centre of the tube bank.

With large quantities of exhaust steam it is very important that every square metre of cooling surface is active and large steam lanes are required. Fig. 18 shows the tube arrangement for a large condenser, the individual tube banks are of radial flow design with air coolers and air extraction in the central part of the tube banks.



Large Condenser with Radial Flow Tube Banks

Fig. 18

The Jet Condenser

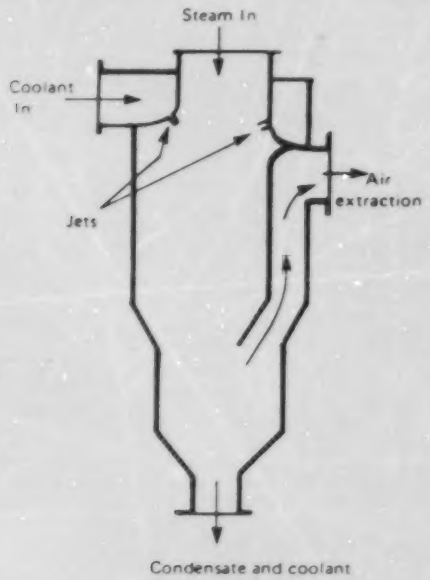
This type of condenser uses the principle of spraying cooling water into direct contact with the incoming exhaust steam in order to condense it. The combined cooling water ( coolant ) and condensed steam ( condensate ) is then taken to the hotwell.

The feed pump draws from this hotwell only the quantity of feed required by the boilers, the remainder overflows into the cooling pond where it is cooled and returned to the condenser again as cooling water.

This arrangement has the definite disadvantage that the entire cooling water quantity must be chemically treated to maintain boiler feed water purity, and for this and other reasons there are very few jet condensers in operation today.

( PE3-4-2-13 )

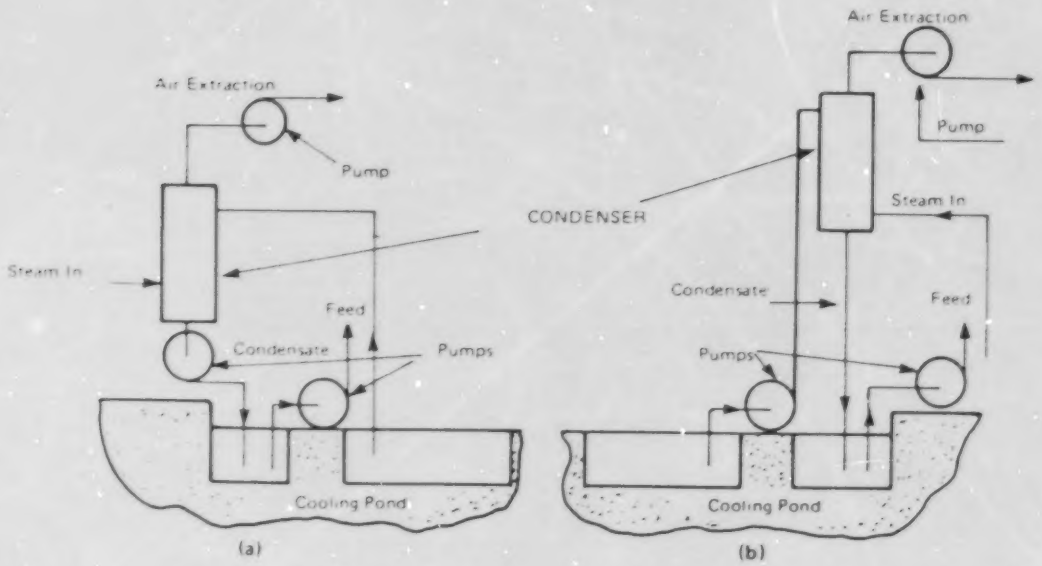
Fig. 19 shows diagrammatically the operation of a jet condenser.




Jet Condenser

Fig. 19

Fig. 20 ( a ) and ( b ) show two methods of application of the jet condenser.




Jet Condenser Applications

Fig. 20

( PE3-4-2-14 )

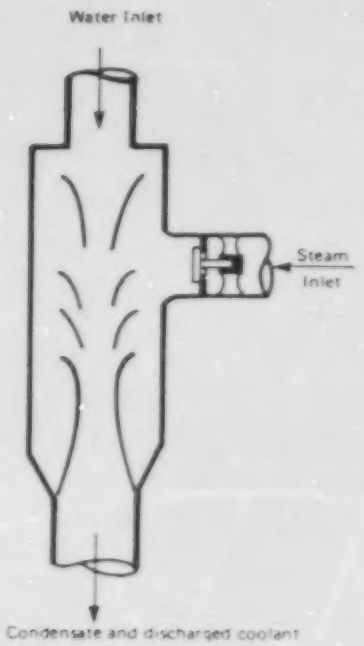
In Fig. 20 ( a ) the condenser is set at a low level, the air and condensate are pumped from the condenser and the cooling water is induced to flow in by vacuum.

In Fig. 20 ( b ) the condenser is set about 35 feet above the level of the hotwell so that the head of water ( condensate and coolant ) in the discharge pipe is sufficient to cause it to fall into the hotwell against the vacuum in the condenser, without pumping. This is a so-called "barometric leg". The cooling water must be pumped into the condenser and the air pumped out.

Finally, the ejector condenser is a special form of jet condenser.

The cooling water flows into the condenser body through a series of convergent nozzles which increase its velocity, steam from the engine exhaust is induced into this cooling water flow and condensed by direct mixing.

The ejector condenser is suitable only for moderate vacuum. Fig. 21 illustrates the ejector condenser.




Ejector Condenser

Fig. 21

( PE3-4-2-15 )

Condenser Auxiliaries

It is impossible to avoid small quantities of air entering a steam turbine system in the areas which are under vacuum. These small air quantities accumulate in the steam space of the condenser and must be removed continuously as the absolute pressure in the condenser according to Dalton's Law is the sum of all partial pressures or the sum of the steam pressure and the air pressure; thus the air increases the condenser pressure and may air blanket part of the cooling surface.

The usual means of extracting the air is by means of an air ejector, although in some cases an air pump, either reciprocating or rotary, may be used for this purpose.

1. The Air Ejector

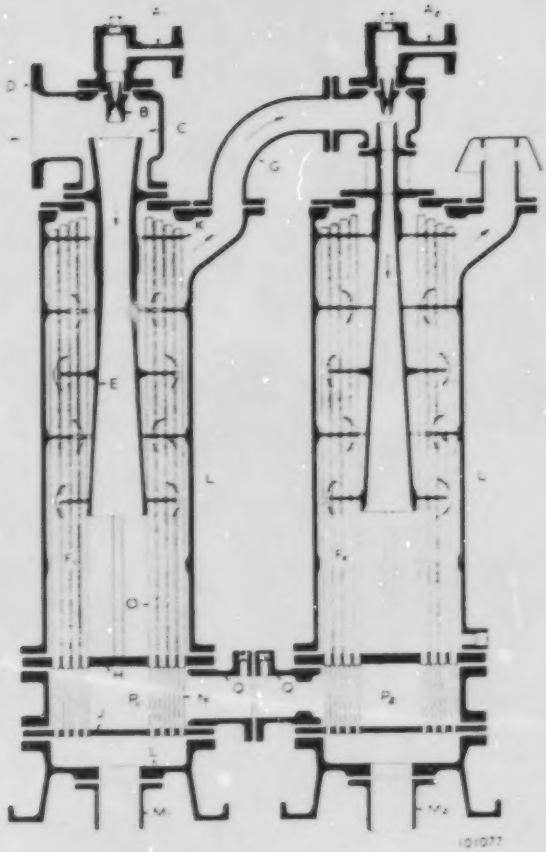
In the ejector, high pressure steam is allowed to expand through a nozzle, thus converting its heat energy into kinetic energy and producing a high velocity jet at the nozzle discharge. This jet is used to entrain air and other non-condensibles and draw them from the condenser space. Most condensers have baffle plates to guide the air to special air coolers at the air extraction points.

Fig. 22 shows a two-stage steam operated air ejector. High pressure steam up to 3000 kPa is supplied to both nozzles and allowed to expand through a small orifice. The resulting high velocity steam entrains gases from the condenser and carries them into the first stage shell- and-tube condenser. Here the steam is condensed and the remaining non-condensibles are drawn off again by being entrained in the steam jet from the second stage nozzle.

The steam is condensed in this stage but here the pressure is maintained slightly above atmospheric pressure so that the remaining air and gases can be vented to the atmosphere. The condensed steam is drained off to a drains tank.

The pressure in the first stage ejector cooler is below atmospheric but above condenser-pressure and the condensed steam is drained back to the condenser; a loop seal on the drain piping prevent recirculation of non-condensibles.

For high vacuum three-stage ejectors are used and the second-stage ejector cooler condensate is drained via a loop sealed drain pipe to the first-stage.



Section of a two-stage air ejector with inter and after surface coolers.

Fig. 22

The cooling water for the ejector cooler in Fig. 22 is condensate from the extraction pump, it enters at $M_1$, flows through the inner tubes N to the closed top of the outer tubes O, down through the outer tubes to chamber $P_1$ over to chamber $P_2$, up through the outer tubes and down through inner tubes and out at $M_2$. The use of condensate makes the ejector coolers a kind of feedwater heater recovering the heat losses of the jet steam.

The orifice of the ejector nozzle is extremely small and care must be taken to see it does not become choked with foreign matter from the supply steam pipes. A fine mesh steam strainer is usually fitted upstream of the nozzles for this reason. Water ejectors working on the same principle as the ejector condenser ( Fig. 21 ) are often used as air ejectors. The ejector is supplied with cooling water from a booster pump at 600 to 800 kPa.

( PE3-4-2-17 )

## 2. Condenser Safety Fittings

A condenser would not be complete without certain safety fittings, designed to protect both the condenser and the turbine exhausting into it, against possible operating troubles. The main dangers to be guarded against are: an increase in back pressure, a rise in condensate level, and contamination of condensate.

### a) Atmospheric Relief Valve

The condenser is a closed vessel and therefore it would be possible for the back pressure to rise until it was above atmospheric pressure if, say, the cooling water flow stopped while the condenser was on load. A condenser shell is not designed to withstand a pressure from the inside and would soon burst.

The atmospheric relief valve is designed to open if the pressure in the condenser rises above atmospheric and allow the steam to escape from the shell. Under normal operating conditions this valve is held closed by the difference in pressure between the atmosphere outside and the vacuum in the shell. In order to ensure that air does not leak past the valve into the condenser, it is usually fitted with a water seal. It operates on a balanced lever and should be tested at frequent intervals when the machine is off load to ensure that it is quite free.

Note that since the purpose of the atmospheric relief valve is to vent the full flow of exhaust steam to atmosphere it is necessarily of considerable size.

For large condensers the atmospheric relief valve is replaced with explosion diaphragms on all L.P. turbine exhausts. Other protective devices for condenser pressure are: vacuum pay-off relays and vacuum trip relays. The vacuum pay-off relay is incorporated in the turbine governor system and is usually set to operate between 10 kPa and 40 kPa absolute pressure, so that unloading of the turbine begins at 10 kPa and the turbine is fully unloaded at 40 kPa.

The vacuum trip relay is set to trip the turbine at 50 kPa.

### b) Condenser Gage Glass

This gives a clear indication of the level of the condensate in the condenser hotwell. The top and bottom of the glass are connected above and below the water level and the whole of the fitting is under condenser vacuum.

Care must be taken to see that there are no air-leaks in the fitting, particularly through the cocks. A board, painted with diagonal stripes and placed

behind the glass makes the water level easier to read.

c) High Water-Level Alarm

The indication of condensate level given by the gage glass is often supplemented by a float-operated alarm to give warning of high-water level.

A steadily rising condensate level would very quickly seal off the air outlet, the vacuum space would become filled with non-condensible gases, the back pressure of the exhaust steam would increase and the turbine output would fall.

d) Detection of Cooling Water Leaks

If a condenser tube is damaged, or a ferrule begins to leak, cooling water can find its way into the steam space of the condenser and contaminate the condensate. It is most important that this can be recognized at once and corrected. There are several methods employed for this purpose, the most common being an electrical purity measurement.

Cooling water is a very good conductor of electricity because of its impurities, whilst the very pure condensate is a non-conductor. Leakage of cooling water into condensate may therefore be detected by taking readings of the electrical conductivity of a sample of condensate in a dionic tester.

The presence of salt in water can be detected chemically by the silver nitrate test, this method can be used when the cooling water is salt. When the condensate sample contains traces of sodium chloride, the sample will turn milky-white when a few drops of a silver nitrate solution is added. When leakage has been confirmed the repair must be made as soon as possible.

If the condenser is of the single flow type, the condenser must be taken out of service for repair. When the turbine is shut down, the waterboxes are drained and the inspection doors opened.

The steam space can be filled with clean water, but before doing so, the supporting or jack-screws should be applied to carry the weight of the condenser.

With the steam space under water pressure the water will leak out of the leaky tube. However, the leaks detected by the conductivity meter may be very small and approximately 25 grams of fluorescin dye, a yellowish-red crystalline compound may be added. The water from the leaky tube will then show up

fluorescent green in the light of a special detector lamp.

Large condensers are usually of the double flow type with tubes expanded at both ends and have the advantage that leaky tubes may be located and plugged while the unit is at 50% load or less. The procedure is: drain one side of the waterbox, the conductivity meter will indicate if that is the leaky side or not. When the leaky side is detected, it is drained.

It is very important to isolate the air ejector suction on the drained side, because the air ejector will be choked with steam as the air cooler is out of service. With a clear plastic hose hooked up as gage glass for the waterbox, it is filled, say, 25 cm at a time and the conductivity meter is checked each time; when conductivity shows the leak, it should be within 25 cm of the waterlevel and that area will, as soon as the waterbox is drained again, be investigated for leaks by covering the tubes at both ends with thin household plastic wrapping taking a small area, say, 25 x 50 cm at a time. The condenser vacuum will suck the plastic in at the leaky tube, when the plastic bursts the leak is located. The tube is plugged at both ends with wooden plugs and the condenser is brought back into service again.

e) Air Leaks

Air leaks may be quite large, approximately 4 - 5 mm in diameter, before they interfere with the vacuum. The air ejector can usually handle any leak smaller than that. There is in most cases only one way to detect an air leak, that is to shut down the turbine and condenser and fill the low pressure heaters and the condenser steam space right up to the turbine blades with water, and the leaks should not be difficult to locate.

3. Circulating Water ( CW ) Pumps

These pumps, which are also referred to as cooling water pumps, are used to pump water through the tubes in a surface condenser. The water is usually pumped from a river or lake and after passing through the condenser is returned to the river or like once again. In locations where there is a water shortage, the water, after leaving the condenser, is cooled in a cooling tower and returned once again to the condenser for re-use.

A type of pump commonly used for circulating water service is the vertical pump shown in Fig. 23. It may be an axial-flow, also called propeller type pump, where the impeller is the shape of a ship's propeller; or it may be a "mixed-flow" pump obtaining its pumping action from a mixture of centrifugal force and the lifting effect of the impeller vanes. The pump bearings may be oil lubricated bronze or babbitt lined bearings or they may be nylon- or hard rubber-bearings with clean water injected at the centre of the bearings for continuous flushing and lubrication.

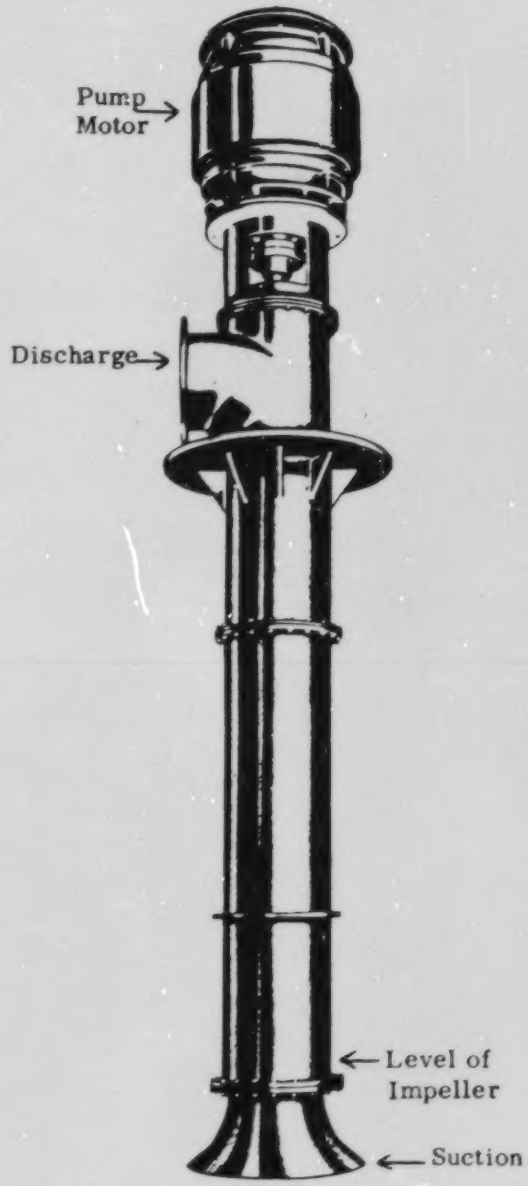
The vertical design makes the pump length variable by adding more sections of pump casing and shaft. This is important as the impeller must be submerged for proper operation and it also eliminates the need for priming the pump.

The axial thrust on the impeller caused by the discharge head together with the weight of impeller, shaft and the rotor of the motor is carried by a large Kingsbury thrust bearing at the top of the motor.

For large condensers the propeller pump has been replaced with large single stage, double suction, vertical centrifugal pumps, a design that eliminates axial thrust.

The same type of pump, but in a small and horizontal version is often used for small condensers.

If not submerged the centrifugal pumps must have an arrangement for priming.




Vertical Mixed Flow CW Pump

Fig. 23

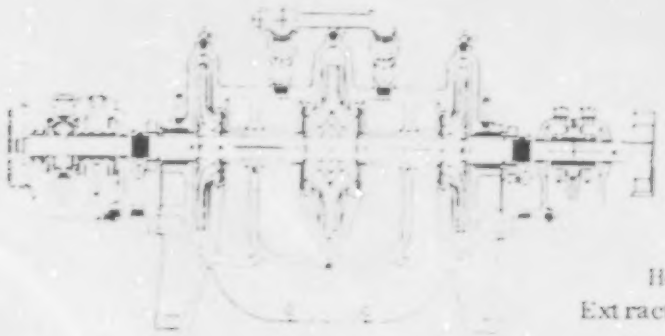
(PE3-4-2-21)

4. Extraction ( Condensate ) Pumps

The function of the extraction or condensate pump is to continually extract the condensate from the condenser hotwell and pump it through air ejector coolers and low pressure heaters into the deaerator.

Except for handling clean condensate, the extraction pumps operate under very severe conditions as the suction pressure is near absolute zero pressure, the condensate is near the boiling point at the entrance to the pump and it must keep nearly constant level in the hotwell at any load on the condenser. It is standard practice to have two 100% extraction pumps for each condenser.

The extraction pump is usually a centrifugal pump with two or three stages. Fig. 24 shows a two-stage horizontal extraction pump, the first stage has a double suction impeller and discharges the condensate in a split stream to the two opposed second-stage impellers working in parallel.



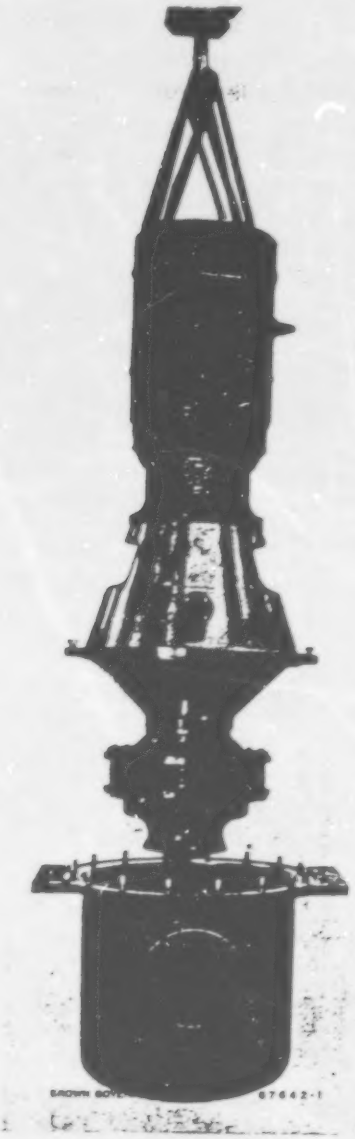
Horizontal Two Stage
Extraction Pump ( Allis-Chalmers )

Fig. 24

The glands are watersealed, to prevent the entrance of air when the pump is on stand-by. In service the glands are exposed to the pump discharge pressure. For the proper operation of an extraction pump, the pump must have an air release connection from the pump discharge back to the condenser, or a small equalizing connection from discharge back to the suction. When the pump during low load loses the condensate, the pressure across the pump will equalize through the equalizing connection, or the air release; and condensate will again be extracted from the condensers as it becomes available for the pump.

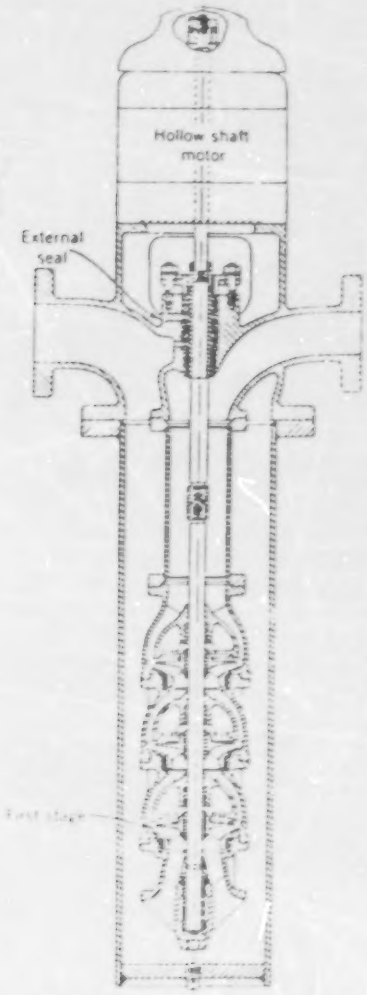
The very low N.P.S.H. of extraction pumps makes cavitation much more common in these pumps than in any other pumps. For reduced cavitation the vertical well type extraction pump ( Fig. 26 ) is preferred. The flange of the well is

level with the floor and the suction head is increased. The well is air-tight with an air-release line leading back to the condenser. These pumps have the advantage of better performance and less cavitation.



Vertical Condensate Pump

Fig. 25

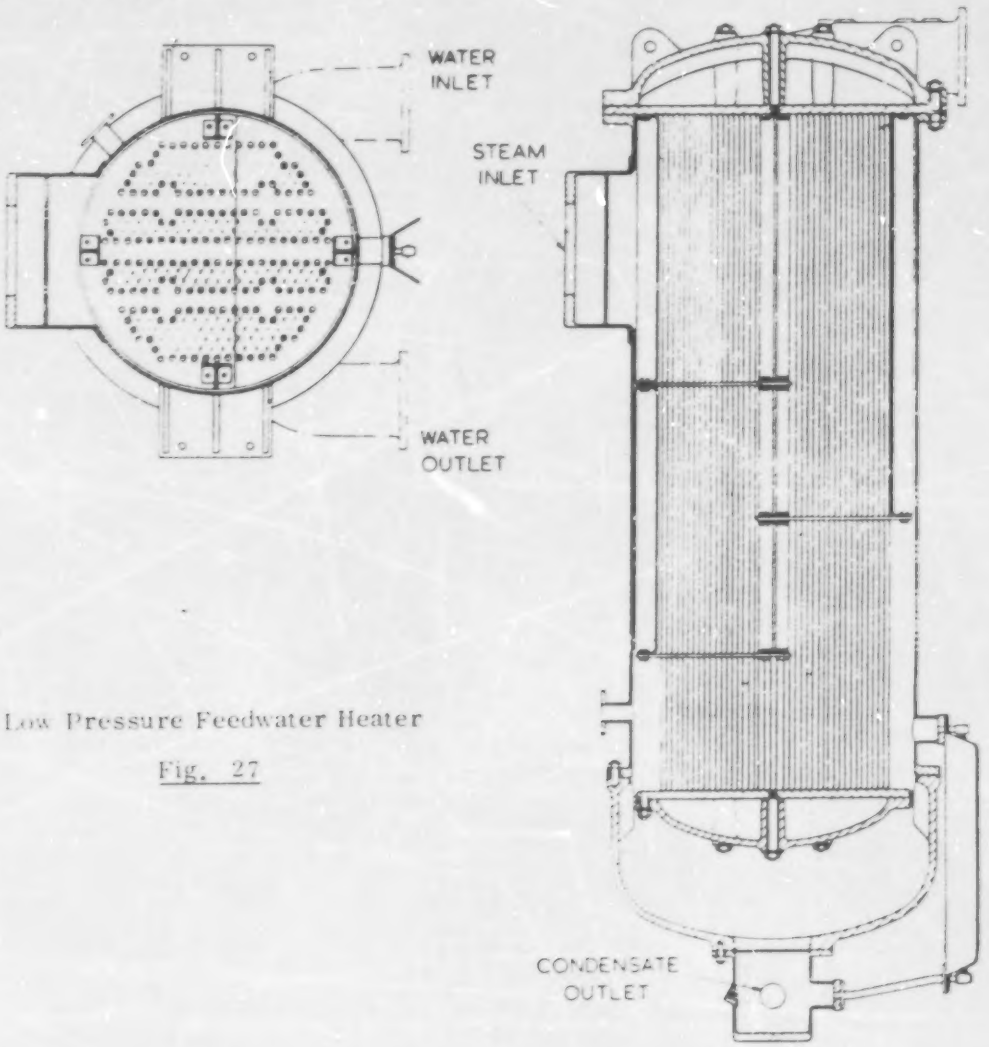



Vertical Multistage Condensate Pumps of the Turbine Type

Fig. 26

## FEEDWATER HEATERS

As explained previously the latent heat of the turbine exhaust steam is lost to the condenser cooling water. The application of feedwater heaters where the latent heat of bleed steam is used for heating feedwater, reduces the quantity of steam exhausted to the condenser and increases the efficiency. The feedwater heaters are divided into low-pressure and high-pressure heaters. The low-pressure heaters are on the water side exposed to the discharge pressure of the extraction pump, while the high-pressure heaters usually are exposed to the boiler feedpump pressure on the water side.



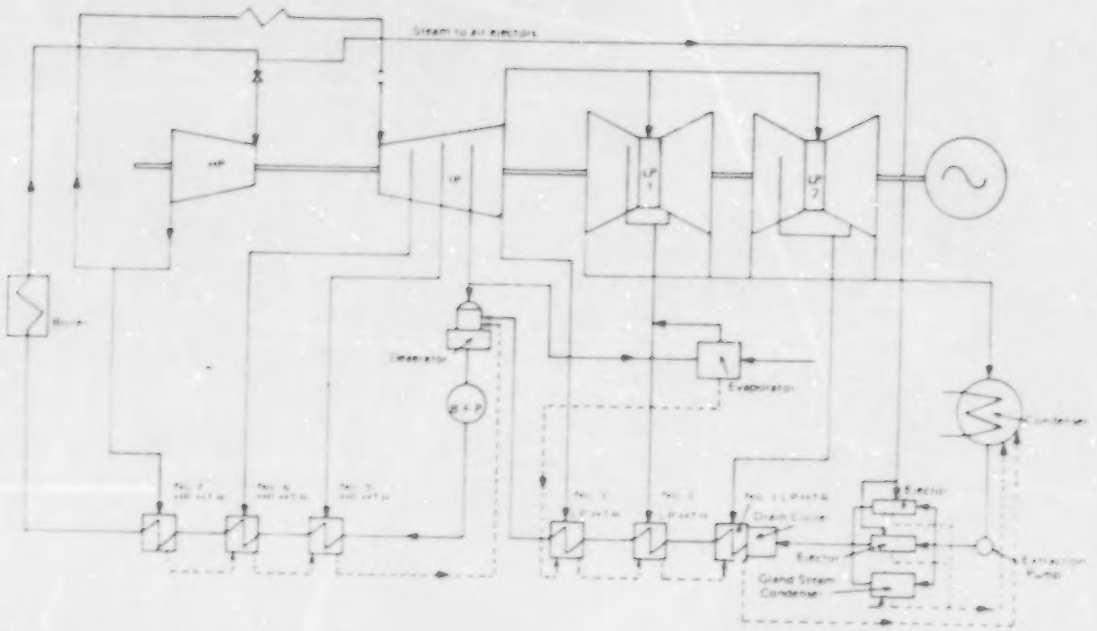

Low Pressure Feedwater Heater

Fig. 27

( PE3-4-2-24 )

Fig. 28 shows the feedwater heater system for a large steam turbine. The condensate passes through the air-ejector coolers, drain-cooler, low pressure heaters into the deaerator heater.

The boiler feed pump then pumps the deaerated water from the deaerator through a series of three high pressure heaters to the boiler. The feedwater heaters, both high pressure and low pressure, and the deaerator are all supplied with steam bled from various stages of the turbine.



Feedwater Heater System

Fig. 28

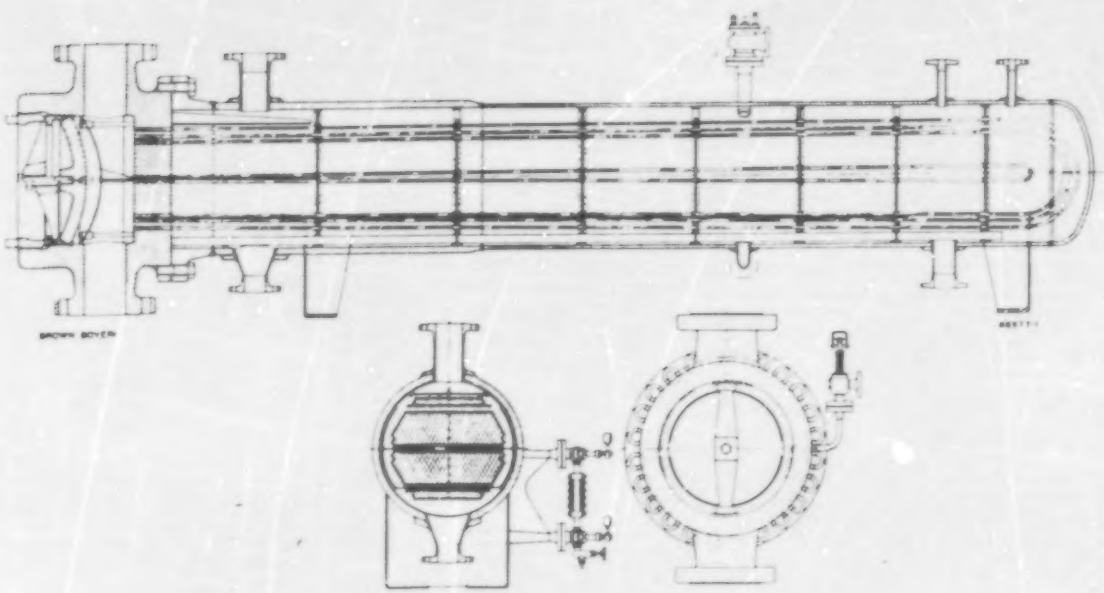
( PE3-4-2-25 )



BEST COPY AVAILABLE

Low pressure heaters are usually of a straight-tube design, tubes and tube-sheets are of brass, the shell is of mild steel, the waterboxes are of steel or cast iron. The shell may have an expansion bellows or a floating waterbox as in Fig. 27 which allows for expansion and contraction.

The boiler feed pump forces the feedwater through the tubes of the high pressure heaters. These tubes are of carbon steel and the "U" tube design is general due to the excellent and simple solution for expansion and contraction. ( Fig. 29 ) The tubes may be expanded or, most common today, welded to the tube plate ( Fig. 30 ).



High Pressure Feedwater Heater

Fig. 29

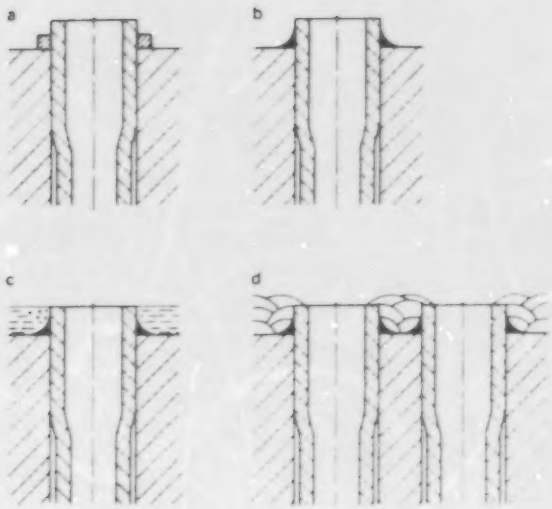



Fig. 30

Stages of tube welding process

a The tubes and the holes in the tube plate are thoroughly cleaned, a ring is placed over the protruding tube ends and the end part of the tube is slightly expanded
b The ring made of special alloy is melted using an automatic argon arc welding gun
c The weld is checked for gas-tightness
d The spaces between the tubes are filled up by hand welding, this is followed by heat treatment and surface treatment

Heater Fittings

For the operation, maintenance and safety of feedwater heaters, the following valves are installed:

Safety Valve on steam side,

Relief Valve on the waterbox to prevent excessive pressure due to thermal expansion of the water when the waterside is isolated.

Bleed Steam Stop Valve and Non-return Valve - the non-return valve prevents steam from the heater entering the turbine, which may occur when the turbine load is suddenly reduced. This causes the bleed steam pressure to decrease and part of the condensate in the feedwater heaters flashes into the steam, which if allowed to enter the turbine, would cause it to increase speed after the overspeed trip has closed the turbine main stop valve.

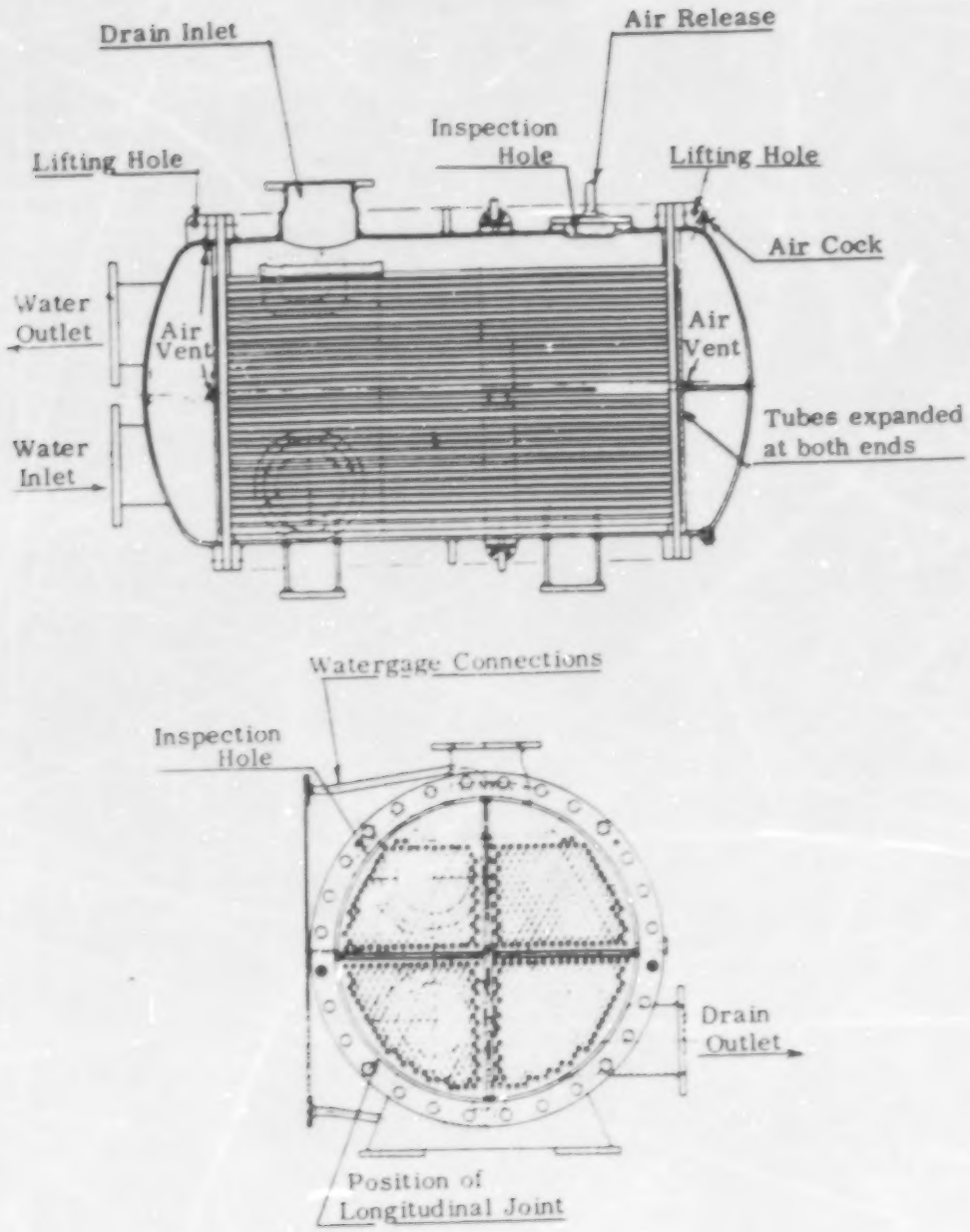
Drain Valve on steam side for draining off condensate.

Drain Valve on water box for draining off waterside for maintenance.

Air-release Valve on steam side for air-release to the condenser, as air trapped in feedwater heaters prevents steam from coming in contact with the air-bound tubes. A low-pressure heater with isolated air-release connections may become completely air-bound and no heat at all will be transferred to the feedwater.

The condensate from the steam space of a feedwater heater passes through an orifice plate in the drain line and drains to the steam space of a heater at lower pressure, see Fig. 28. For vertical heaters the orifice plates provide sufficient water level control for the heaters. The water level is much more critical in horizontal heaters and automatic level control valves are usually required.

If the drain from the first low-pressure heater is allowed to drain back to the condenser, a considerable amount of sensible heat is lost in the condenser. This can be prevented by the installation of a drain cooler ( Fig. 31 ). The condensate leaving the air-ejector coolers flows through the tubes of the drain cooler and most of the sensible heat of the drains is absorbed, before they are allowed to enter the condenser. In many installations a condensate return pump, replacing the drains cooler, returns the condensate from the heater steam space back to the main condensate pipe at the outlet from the low-pressure heater, in which case all the sensible heat from the drains is reclaimed.




- Sectional End Elevation -

Drain Cooler ( C.A.P. Design )
Fig. 31

( PE3-4-2-29 )

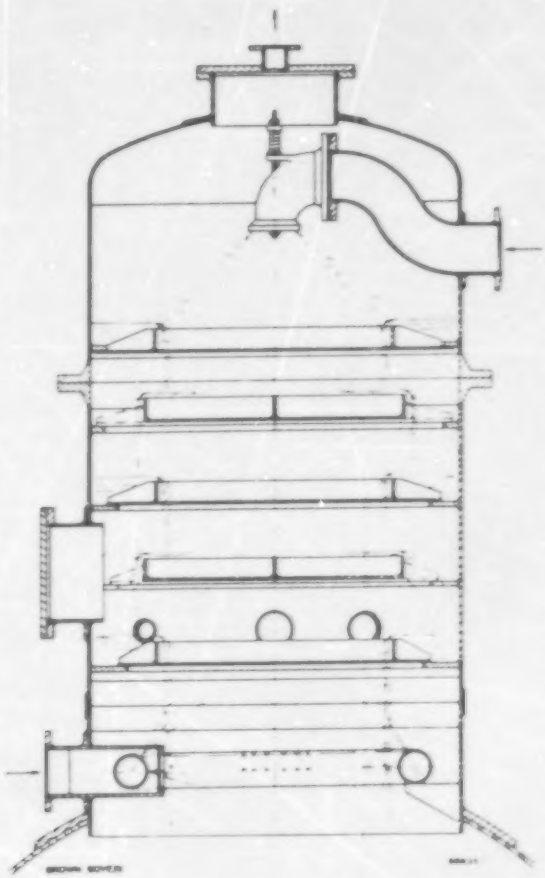
## DEAERATORS

The deaerator or deaerator-heater serves a dual purpose as it heats and deaerates the condensate.

Thermal deaeration is accomplished by heating the condensate to the boiling point liberating all dissolved gases. The deaerator must be designed to heat the condensate to the boiling point at the rate at which it is pumped by the extraction pump. This is usually accomplished by mixing steam and condensate. A mixing-deaerator of simple construction is shown in Fig. 32. At the top of the deaerator the condensate flows through a nozzle into the top of the steam chamber, where it mixes with bleed steam increasing the temperature to near the boiling point. The water cascades in a thin sheet down over a series of trays, where most of the deaerating takes place.

The deaerator is usually located on top of the feedwater tank, ( deaerator and tank are generally considered as a unit and called the deaerator ). The deaerated condensate flows down into the tank while the non-condensible gases flow to the top, where they are vented off to the main condenser.

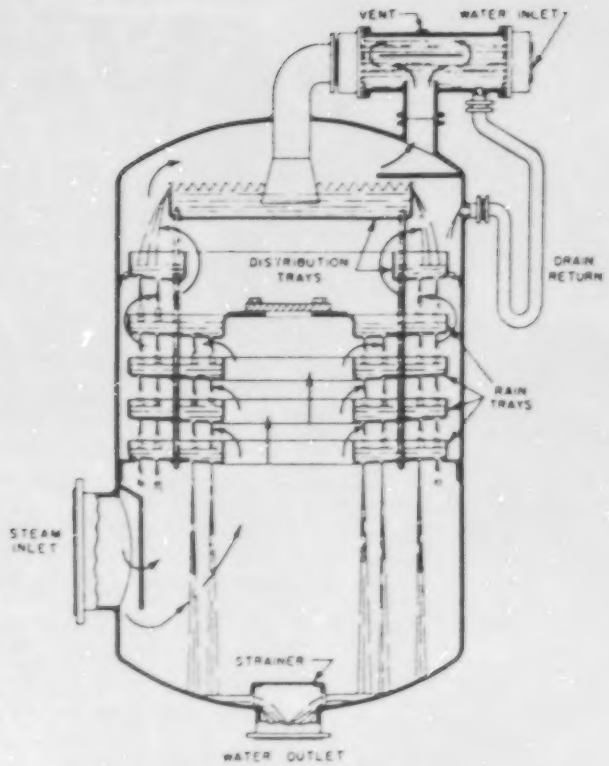
It is very common to install a vent or vapor condenser on the top of the deaerator as shown in Fig. 33.



Brown Boveri Deaerating Mixer Heater

Fig. 32

( PE3-4-2-30 )




Tray-type Direct-contact Heater
( Foster Wheeler Corp. )

Fig. 33

The condensate from the L.P. heater flows through the tubes of the vent condenser before entering the deaerator. The non-condensibles and water vapors enter the vent condenser and flow around the tubes, the vapors condense and return, through a loop-sealed drain pipe,to the deaerator. The loop-seal prevents recirculation of non-condensibles. The non-condensibles are drawn off from the top of the vent condenser to the main condenser. The vent condenser reclaims the latent and sensible heat of the vapors.

The non-return valve in the bleed steam piping to the deaerator prevents the deaerator from acting as a steam accumulator supplying the turbine with steam after a sudden load reduction.

( PE3-4-2-31 )

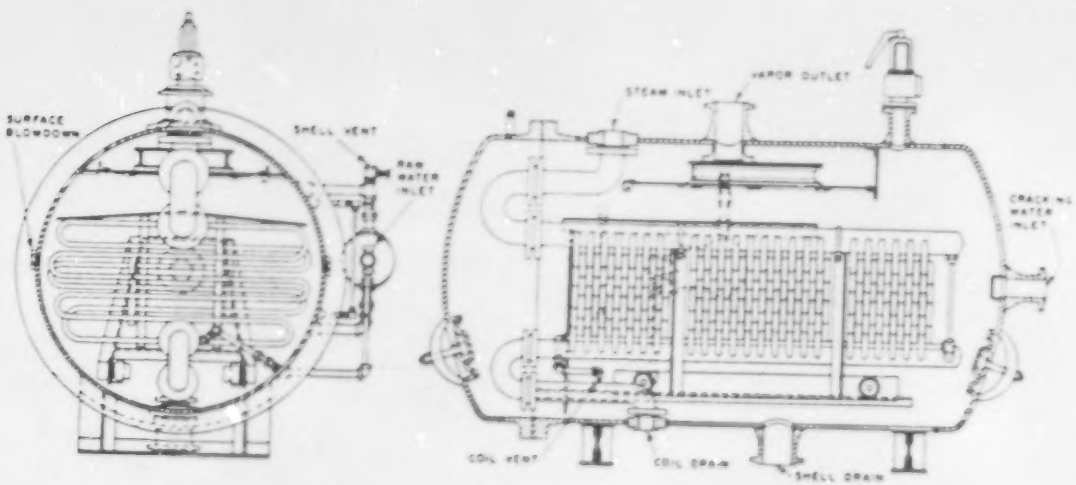
The deaerator tank usually has a low level float switch, which at low level opens a control valve allowing water to flow from a reserve feed tank into the condenser for deaerating before it is pumped to the deaerator by the extraction pump. A high level float switch opens a control valve allowing condensate to by-pass the deaerator and flow to the reserve feedwater tank.

## EVAPORATORS

Most high pressure boilers require pure feed water, either demineralized water or distilled water. The make-up water, replacing the water losses due to soot-blowing, atomizing steam for oil burners, blow-down from boilers etc., must be pure water.

The most effective way of producing pure water is by bleed steam evaporators, where steam bled from the turbine at 120 kPa - 250 kPa is applied to the evaporator steam coils, and the steam or vapor produced by the evaporator is condensed in a low-pressure feedwater heater operating as an evaporator condenser; see Fig. 28. Practically all heat from the bleed steam is recovered except for the heat losses due to blow-down of the evaporator.

A submerged coil evaporator for high quality water is shown in Fig. 34.




Submerged-coil Evaporator
( Foster Wheeler Corp.)
Fig. 34

( PE3-4-2-32 )



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The shell is made of fabricated steel, the rear end of the shell is closed by a welded-on formed head, the front is open with a welded-on hub-type forged flange. The front cover is a formed head with a welded-on heavy flange. The large front cover bolted to the shell allows for removal of the entire coil and header assembly which is supported on a steel frame equipped with flanged wheels rolling on tracks fastened inside the shell.

The steam coils are all in parallel and fastened to the top and bottom headers which allows for free draining. The coils have several loops making them very flexible allowing descaling by cracking.

Baffles on top of the coils and under the vapor outlet separate water from the vapor. This evaporator is designed for operating with raw water, but treated or softened water is often used, depending on the quality of the raw water and the required quality of the distilled water.

As water evaporates, the dissolved solids are left behind increasing the total dissolved solids in the water inside the evaporator. The evaporator must be blown down frequently in order to keep the dissolved solids inside limits, to prevent foaming and carry-over.

As a pressure vessel the evaporator must have a safety valve. The bleed steam inlet must have a stop and non-return valve, and the coil drain should have steam traps or for large evaporators water level control valves. A float control valve maintains the water level and a gage glass is installed directly on to the shell and indicates the water level.

Good heat transfer requires clean heating surfaces. A fast and effective cleaning method is cracking where the evaporator is drained and steam is admitted to the coils. The expansion of the hot tubes may crack some of the scale. Then the steam supply is shut off and cold water is sprayed over the tubes. The contraction of the tubes cracks off more scale. Repeating the cracking procedure several times takes care of most of the scale, but every so often the coil assembly must be taken out for complete cleaning of the coils as well as the inside of the shell.

## COOLING TOWERS AND PONDS

If a sufficient and suitable supply of cooling water for the condenser is not available, then some means must be used to re-cool any water that is available so that it can be used over and over again.

To achieve this, either cooling towers or cooling ponds are used and in both methods the warmed water is cooled by exposure to the atmosphere and subsequent evaporation of a portion of the water.

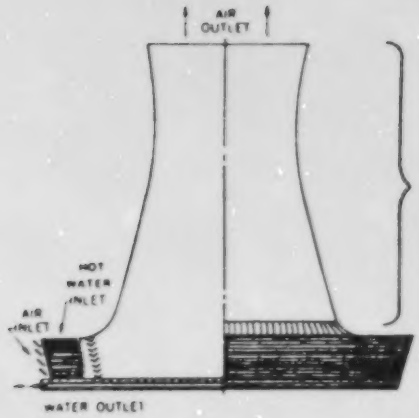
In the case of a cooling pond, the warm water is discharged to a pond of sufficient area to allow it to be cooled by contact with the air at the surface of the pond. If the water is sprayed into the air above the pond surface, then the pond area can be considerably reduced as the spraying will provide additional contact between water and air.

In the case of a cooling tower, the warm water is delivered to the top of the tower and falls by gravity over an arrangement of splash bars to a reservoir at the bottom. The air is caused to rise up through the falling water by natural draft or by mechanical draft. The natural draft type uses a chimney or stack to induce a flow or air through the tower while the mechanical draft type uses a fan to cause the air flow.

### Natural Draft Tower ( Hyperbolic Type )

The hyperbolic natural draft cooling tower consists of a lower and an upper portion. The lower portion is the cooling section which contains wooden splash bars over which the warm water falls as it travels to a collecting basin or reservoir directly below the splash bar section. The air enters around the perimeter of the splash bar section. The upper portion of the tower is a hyperbolic shaped section constructed of reinforced concrete. This section acts as a chimney to induce the flow of air through the cooling section.

The hyperbolic tower has the advantage of high capacity, low power consumption, long service life and a minimum of maintenance. It does, however, have a high initial cost. The sketch in Fig. 35 shows the general arrangement of a hyperbolic tower.




Hyperbolic Tower

Fig. 35

## Mechanical Draft Towers

The mechanical draft tower may be either a forced draft type, or an induced draft type.

In the forced draft type, the air for cooling is forced through the tower by a fan which is located at the base of the tower.

In the induced draft type, the fan is located at the top of the tower and draws air in through louvres situated around the tower base. The air passes upwardly through the falling water and then passes through baffle type drift eliminators which separate any entrained moisture from the air. The air is then discharged at a relatively high velocity by the fan.

The induced draft tower is more commonly used than the forced draft type as it has the following advantages:

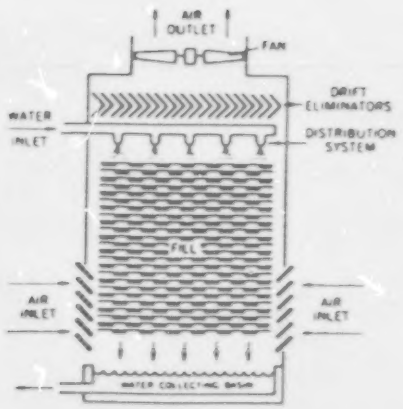
1. As the air is discharged at high velocity, it is not likely to recirculate back to the inlet around the tower base.

2. The fan is not likely to ice up as it is in the path of the warmed air being drawn from the tower.

3. The noise of the fan is not so noticeable as it is located at the top of the tower rather than at the bottom.

An induced draft tower arrangement is sketched in Fig. 36.

( PE3-4-2-35 )




Induced Draft Tower

Fig. 36

## STARTING AND STOPPING STEAM TURBINES

In the following sections the general methods of starting and stopping non-condensing and condensing steam turbines will be discussed. It is not possible to give detailed instructions because of the large number of different makes and designs of these machines, each of which differs in some detail from the others. Therefore the plant personnel who are required to operate the particular turbine should consult and become familiar with the instruction books issued by the manufacturer of the machine.

### Starting a Small Non-condensing Turbine

1. Check that governor linkages are well lubricated and grease cups if used are filled. Check oil level in bearing sumps if bearings are ring oiled or check level in main oil reservoir if a pressure lubricated system is used.
2. Make sure that all condensate is drained from steam lines and exhaust lines. If these are normally drained by means of traps then open by-pass lines around traps.
3. Slowly open the shut-off valve between the turbine and the exhaust line.
4. If the turbine is equipped with a pressure lubricated system having a separately driven oil pump, then start the pump. Check that sufficient oil pressure and flow is established.

5. If bearing oil sumps or oil reservoirs are water cooled, then turn on water supply to them.

6. Partly open the throttle valve quickly to start the turbine rotor turning. Then close the valve again to the point where the rotor continues to turn slowly. Listen carefully for any noise that indicates rubbing of the rotor against the casing. In case of ring-lubricated bearings check the operation of all lubricating rings.

7. If no unusual noises are detected, increase the turbine speed to between 200 and 300 rev/min. Maintain this speed for about one half hour to allow rotor and casing to reach operating temperature.

8. Trip the overspeed valve by hand to check its operation.

9. Restart the turbine as in Item 6 and bring turbine up to rated speed. Check that as the machine approaches this rated speed, the governor takes over control and maintains this speed. The throttle valve may now be opened wide.

10. If a main oil pump driven by the turbine is installed then the separately driven auxiliary oil pump may now be shut down. Check oil pressure and flow once again.

11. Steam line drains and trap by-passes may now be closed.

12. Increase load on turbine gradually.

Stopping a Small Non-condensing Turbine

1. Reduce turbine load to zero.

2. If a separately driven auxiliary oil pump is used, start this pump.

3. Shut off steam supply to the turbine by manually tripping the over-speed trip valve. Then shut the throttle valve.

4. Shut off water supply to oil coolers.

5. When turbine comes to rest, shut the exhaust line valve.

Starting a Large Condensing Turbine

1. Two to three hours before steam will be admitted to the turbine start the lubricating oil pump. The pump supplies lubricating oil at 75 to 100 kPa to the bearings.
2. Start the jacking oil pump which supplies lubricating oil at 8000 - 10 000 kPa to the bottom of the bearings in order to float the shaft.
3. Engage and start the barring gear. The barring gear is usually electrically interlocked by a lubricating oil pressure switch and and engagement switch.
4. When the barring gear is up to speed, shut down the jacking oil pump.
5. When the turbine has been barring for two to three hours, open drain valves on steam line and turbine casing and open trap by-passes.
6. Start auxiliary oil pump. If it is automatic, turn selector switch to "auto ' to start the pump. This is a separately driven oil pump which supplies lubricating oil at 75 - 100 kPa and governor oil at 550 - 650 kPa during start-up and shut-down periods.

   When the turbine is in normal operation this oil is supplied by the main oil pump, which is driven by the turbine shaft. The main oil pump pressure is set slightly higher than the auxiliary oil pump pressure in order to make an oil pressure switch shut off the auxiliary oil pump when the main pump takes over.
7. Shut down lubricating oil pump. If automatic it should shut down when the auxiliary oil pump takes over.
8. After starting the auxiliary oil pump, check the oil pressure produced, the oil flow to each bearing and the oil reservoir level.
9. Start the circulating water pump and establish circulating water flow through the condenser.
10. Start the extraction ( condensate ) pump and circulate condensate through the air ejectors.
11. Admit sealing steam to the turbine glands.
12. Put the air ejector into operation and draw a partial vacuum on the condenser.

( PE3-4-2-38 )

13. Open the throttle or steam admission valve just enough to start the turbine rolling.

14. Trip the overspeed valve by hand to see that it is free to operate.

15. Restart turbine as in Item 13 and maintain it at a low speed while checking around the turbine listening and looking for anything unusual.

16. Gradually admit more steam to turbine in order to slowly increase the speed. If any unexpected vibration occurs then decrease speed again and continue warming up until turbine runs smoothly when the speed is next increased.

17. Check that the governor takes over control when rated speed is reached.

18. In some machines there is a manual over-ride on the governor that can be used to actually overspeed the turbine in order to test the overspeed trip mechanism. If this is the case, the overspeed trip should be tested by this method at this point.

19. Reset the trip mechanism and bring the speed back up to normal once again.

20. When the oil has reached the normal operating temperature, admit cooling water to oil coolers to maintain desired oil temperature.

21. Stop the auxiliary oil pump and check that the main oil pump is maintaining correct oil pressure.

22. The steam admission valve may now be opened fully and the turbine load slowly increased to the desired point.

23. Drain valves may now be closed.

Stopping a Large Condensing Turbine

1. Reduce turbine load gradually to zero.

2. Shut off steam to turbine by manually tripping the overspeed trip, or by closing the steam stop valve manually.

3. Shut down the air ejectors and break the condenser vacuum.

4. Shut off cooling water to oil coolers. Sub-cooling of lubricating oil should be avoided.

5. As the turbine speed decreases, see that the auxiliary oil pump starts automatically at the correct point. If not automatic, start the auxiliary oil pump when the turbine speed is down to approximately 75%.
6. Shut off gland steam.
7. Shut down the extraction pump.
8. Open the steam drain valves.
9. When the turbine comes to a standstill, switch the auxiliary oil pump switch to "off" and the lubricating oil pump should start automatically.
10. Engage the barring gear.
11. Start the jacking oil pump.
12. Start the barring gear.
13. When the barring gear is up to normal barring speed shut down the jacking oil pump.
14. When the machine is cooled off, stop the circulating water pump.
15. Keep the turbine barring for 24 hours, then shut down the barring gear, it should disengage automatically, but check for proper disengagement.
16. Shut off the lubricating oil pump.

QUESTION SHEET

Power Engineering Department

Third Class
Sect. 4, Lect. 2

1. a) Explain the purpose of an evaporator and describe how this purpose is achieved.

   b) What causes high concentration of solids in an evaporator and how is this concentration controlled?

2. Sketch a typical deaerator and describe its operation.

3. Sketch a typical low pressure or high pressure feedwater heater. Include all necessary valves and fittings.

4. List three purposes of a surface condenser and explain how each purpose is achieved.

5. Sketch and describe the operation of a two-stage steam operated air ejector, including ejector coolers with condensate drains.

6. Describe one method of locating a leaking condenser tube.

7. List three operating troubles of a condenser and explain why each is undesirable and how each may be guarded against.

8. Explain how and why condenser pressure will vary with changes in turbine load.

9. Describe the principle of operation of a direct contact condenser and give a disadvantage of this type.

10. For a steam turbine, with which you are familiar, list the important data, such as: manufacturer, size, type, rpm, inlet and exhaust steam pressure; then describe the method of starting and stopping the turbine.

# 8.4

STEAM TURBINES -- OPERATION AND MAINTENANCE

## Goal:

The apprentice will be able to describe steps in operating and maintaining steam turbine equipment.

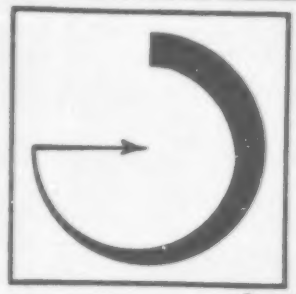
## Performance Indicators:

1. Describe starting procedures.
2. Describe operational procedures.
3. Describe stopping or shut-down procedures.
4. Describe routine maintenance.
5. Describe instruments, controls and supervisory equipment.

# Study Guide

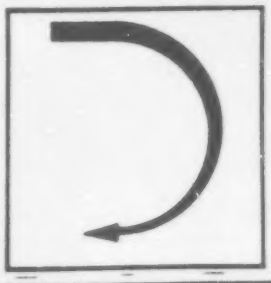
* Read the goal and performance indicators to find what is to be learned from package.
* Read the vocabulary list to find new words that will be used in package.
* Read the introduction and information sheets.
* Complete the job sheet.
* Complete self-assessment.
* Complete post-assessment.

# Vocabulary



* Alignment
* Blade fouling
* Bridge gauge
* Clearances
* Diaphragms
* Glands
* Horizontal eccentricity
* Kenotometer
* Supervisory equipment
* Vertical eccentricity

# Introduction



Steam turbine equipment is constructed in many types and configurations. Each assembly will have unique features. The manufacturers instruction manual should be followed in the operation and maintenance of steam turbine equipment. Safe operation depends on following the recommended procedures in a sequential manner.

This package will describe the general procedures for operating and maintaining steam turbines. Apprentices should learn the general procedures and then turn to operator manuals for information on the operation of specific turbines.

# Information

## STARTING A TURBINE

### New Turbines

When starting a new turbine for the first time, the operator should take great care to see that the internal parts of the turbine are clean. Feedpipes, condensers and other equipment should be cleaned to keep dirt out of the turbine blades. The system should be well lubricated before it is started for the first time. All safety devices, gauges and seals must be inspected or tested to assure that they are functional.

### Starting Instructions

1. Check the condensing plant. Start the circulating pump. Make sure that pipes and water boxes are full of water and clear of air.
2. Check lubrication system. Start auxiliary oil pump and check bearings and valves for oiling. Start jacking oil pump.
3. Engage turning gear to run rotors for warm-up period.
4. Set drains on turbine and steam lines. Open by-pass steam valves.
5. Seal turbine shaft glands and build up vacuum in condensing plant.
6. Start condensate pump and open recirculating valve on condenser.
7. Test emergency trip gear. Shut stop valves and by-passes and open emergency stop valves and test with trip gear.
8. Open stop valve enough to start engine rolling.
9. Close down the turning gear.
10. Bring turbine up to running speed.

# Information

## OPERATING A TURBINE

### Operational Procedures

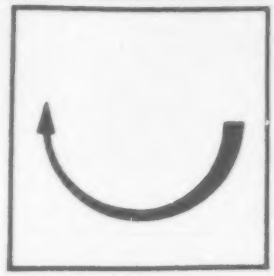
1. Test overspeed trip after governor has taken over control.
2. Check the following before loading:
   - Bearing oil pressures and temperatures
   - Condenser vacuum
   - Steam drains
   - Condensate recirculating valve
   - Thrust adjustment on spindle
   - Auxiliaries such as feed pumps and extraction pumps
3. Keep careful watch on bearing temperatures, vibration and noise during loading.
4. Engage lubricating oil coolers as needed to control temperature of turbine bearings at or near 50 C.
5. Monitor the turbine under load for temperature and pressure of bearings, noises and vibrations. The operator should have some standards for normal operation with which to compare readings. Under load, the turbine pressures will be constant for that specific load. Experience will enable operators to recognize changes from the normal operating temperatures and pressures.
6. Check the back pressure (exhaust vacuum) by use of a kenotometer.

## SHUTTING DOWN A TURBINE

### Stopping the Turbine

1. Set thrust-adjusting gear for maximum clearance.
2. Open condensate recirculating valves.
3. Open main alternator switch after all load has been removed.

# Information



4. Check overspeed trips.
5. Close stop valves on turbine.
6. Shut down air ejectors to destroy vacuum.
7. Open all turbine drains.
8. Check the auxiliary oil pump to see that it starts to operate as turbine speed is decreased.
9. Engage the turning gear.
10. Shut off cooling water valves to the oil coolers to retain heat in the oil.
11. Shut off extraction pumps and circulating water to the condenser.

## MAINTENANCE OF TURBINES

### Blade Fouling

Turbine blades must be kept clean and free of dirt and scale deposits. The cleaning may be done by washing or mechanical means. The turbine can be washed by forcing wet steam through the stop valves. With the cylinder drains open, the operator can determine when the purity of the liquid drain indicates a clean turbine. Washing will not remove scale deposits. A mechanical cleaning must be used to remove insoluble materials. This material can be removed by blasting the surfaces with an abrasive material. A complete cleaning should take place during overhaul while the turbine is dismantled. During overhaul, the blades should be inspected for erosion and cracks. Damaged blades should be repaired or replaced.

### Glands

The operator can detect problems in the shaft glands by the amount of steam required for sealing. During overhaul, the gland packings must be cleaned, adjusted for clearance or straightened as needed.

# Information

Diaphragms

Diaphragms should be inspected for cracks, distortion, rubbing and fit. Nozzles should be cleaned and dressed.

Alignment

The alignment of turbine equipment should be checked if vibration is present. An alignment gauge is used to determine if alignment is correct.

Clearances

Efficient turbine operation requires that correct clearances be maintained between fixed and moving parts of the turbine. If the clearance is too great, steam power is lost. Rubbing and wear of parts occurs if the clearance is too little. When blades, nozzles or packing rings are replaced, the operator must carefully check the clearances. The manufacturer manual will specify the correct clearances for maximum efficiency of the turbine.

Bearings

Maintenance of bearings is critical to the successful operation of turbines. Bearings should be inspected for wear, grooving and electrolysis. The bearings should be checked for their fit and tightness and adjusted when necessary. The oil orifices and passages should be checked to see if they are open. Clearances of bearings should be measured with a bridge gauge and compared with recorded measurements. Changes in bearing measurements show the amount of wear. If wear exceeds the permissable clearance, adjustments or replacements must be made.

## INSTRUMENTS, CONTROLS AND SUPERVISORY EQUIPMENT

Controls and Instruments

Some of the typical controls of a turbine include:

1. Wattmeter to measure load.
2. Pressure gauges to measure:
   a. Steam pressure

# Information

b. Gland steam leakoff
c. Bled steam pressures
d. Aerator pressure
e. Exhaust pressure
f. Bearing oil pressure
g. Relay oil pressure for governor

3. Kenotometer for measuring absolute pressure in condenser.

4. Diaphragm gauges for measuring sub-atmospheric pressure.

5. Deflectional instruments to show levels of water in reserve tank and aerator.

6. Temperature measuring and recording instruments for steam and feedwater.

7. Dissolved oxygen meters and hydrogen ion concentration meters for showing condensate purity.

8. Supervisory instruments.

9. Indicators and alarms that show malfunctions.

Supervisory Equipment

Supervisory equipment is triggered by electronic signals and is used to detect problems caused by excess vibration. Such equipment indicates and records:

1. Vertical eccentricity

2. Horizontal eccentricity

3. Differential expansion

4. Shaft speed

Supervisory equipment is valuable to the operator in starting up, operating and maintaning a turbine. It indicates the condition of the rotor and allows the operator to correct vibration problems before they get out of hand.

# Assignment

* Read pages 17 - 30 in supplementary reference.
* Complete job sheet.
* Complete self-assessment and check answer with answer sheet.
* Complete post-assessment and ask instructor to check your answers.

# Job Sheet

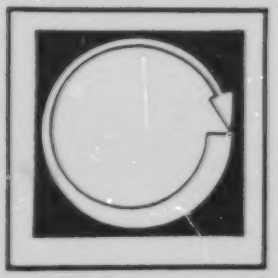
INSPECT A TURBINE CONTROL PANEL

* Locate a site that has a turbine control panel.
* Carefully inspect each dial and control instrument and record on the following instrument.

| CONTROL INSTRUMENT OR SUPERVISORY EQUIPMENT | PURPOSE | UNIT OF MEASUREMENT |
|---|---|---|
| | | |
| | | |
| | | |
| | | |
| | | |
| | | |
| | | |
| | | |

* Ask operator to explain purpose for those dials, gauges, controls that were not included in this package.

# Self Assessment



Show the proper sequence of events for starting a turbine. Show the proper sequence in the spaces at right by numbering in order 1 thru 10.

1. Start condensate pump and open recirculating valve. _____
2. Set drains on turbine and steam lines. Open by-pass steam lines. _____
3. Close down turning gear. _____
4. Test emergency trip gear. _____
5. Check lubrication system. _____
6. Check condensing plant. Start circulating pump and make sure boxes are full of water and clear of air. _____
7. Engage turning gear for warm-up period. _____
8. Bring turbine up to running speed. _____
9. Seal turbine shaft glands and build up vacuum in condensing plant. _____
10. Open stop valve enough to start engine rolling. _____

# Self Assessment Answers

1. 6
2. 4
3. 9
4. 7
5. 2
6. 1
7. 3
8. 10
9. 5
10. 8

# Post Assessment

Match the following terms with their related phrases.

| | | |
|---|---|---|
| ____ | 1. Caused by scale deposits on turbine blades. | A. Kenotometer |
| ____ | 2. Problems can be detected by the amount of steam required for sealing. | B. Deflectional instruments |
| ____ | 3. Should be checked if vibration is present. | C. Dissolved oxygen meter |
| ____ | 4. Used to check clearances on bearings. | D. Alignment |
| ____ | 5. Used to measure purity of condensate. | E. Bridge gauge |
| ____ | 6. Used to measure sub-atmospheric pressure. | F. Diaphragm gauges |
| ____ | 7. Indicates and records eccentricity and expansion. | G. Glands |
| ____ | 8. Measures absolute pressure in condenser. | H. Blade fouling |
| ____ | 9. Shows levels of water in reserve tank and aerator. | I. Wattmeter |
| ____ | 10. Measures electrical load. | J. Supervisory equipment |

# Instructor Post Assessment Answers

H 1.

G 2.

D 3.

E 4.

C 5.

F 6.

J 7.

A 8.

B 9.

I 10.

# Supplementary References

* Correspondence Course. Power Engineering. Lecture 6, Section 3, Second Class. Southern Alberta Institute of Technology. Calgary, Alberta, Canada.

Correspondence Courses
Power Engineering

SECTION 3

PRIME MOVERS

Second Class
Lecture 6

## STEAM TURBINE
## OPERATION and MAINTENANCE

This lecture will deal with operational procedures of running plants. However a few remarks on the commissioning of new plants will not be out of place at this time.

### Cleaning after Erection

On completion of erection of a new boiler and turbine-generator set, elaborate precautions must be taken to ensure the complete internal cleanliness of the entire system.

Considerable operational troubles in steam turbines can be caused by dirt, debris, scale and silica being carried into blading, glands and bearings from the boiler and piping systems.

Earlier lectures have dealt with the cleaning of new boilers, the internal cleaning of the pipework together with the feed-heating and condensing systems will be carried out on somewhat similar lines. The following steps will be taken after the plant erection is complete.

All accessible parts should be well cleaned and all loose material removed. The lubricating and control oil systems should be thoroughly cleaned and then closed. The feed heating and condensing system should be thoroughly hosed and flushed to waste and the system then closed up.

Modern plants demand a higher standard of cleanliness than the above methods will achieve, however, so that these steps are generally followed by alkaline cleaning and degreasing of the steam side of the condenser and feed heaters. A recommended cleaning fluid is a solution of about 200 parts per million of caustic soda and 100 parts per million of trisodium phosphate in deionized water at a temperature of 99°C. Each item of plant treated should be connected so as to produce a positive circulation path and the solution pumped through for about four hours followed by a hot deionized water flush.

A new turbine, being operated for the first time requires careful supervision. Steam joints and small pipework will be left unlagged to facilitate detection of leaks. Certain instruments may not have been provided, however it is essential that the protection and control equipment is fitted and tested before any preliminary runs are attempted. Also safe access to all parts of the machine is essential.

As a further safeguard against dirt and debris the main steam line is often disconnected from the turbine, directed outside of the building and steam allowed to blow through.

The strainers in the steam chest and in the oil supply lines should be replaced with fine mesh for the first few weeks of running time.

All auxiliaries should be tested as soon as electrical and steam supplies are available. A trial of the vacuum raising equipment should be carried out before the turbine is first run. The turbine glands should be sealed, vacuum raised using the ejectors, and the turbine and condenser system checked for air leaks.

The turbine must never be started up without adequate insulation lagging on high-pressure steam pipes and on high-pressure cylinders, otherwise distortion may take place. Speed control during the initial runs should be by hand at the machine. All temperature indicators and pressure gages must be fitted and be operational.

After the initial commissioning and drying-out runs the routine operation of the machine will follow a pattern such as described in the following pages.

Turbine operating methods vary slightly according to the particular machines involved and manufacturers usually issue precise instructions for their individual product. However, the following can be taken as a general guide.

Instructions for Starting

Condensing plant - start the circulating water pump and open up the condenser circulating valves. Ensure that all pipes and water boxes are clear of air and full of water.

Lubricating oil - check oil system valves, start auxiliary oil pump and see that all bearings are being supplied at the correct pressure and that the control system oil supply is normal. Start the jacking oil pump to ease the shafts on the bearings.

Turning gear - engage and start the turning or barring gear to run the rotors for a period of time. This period will vary with the temperature of the turbine, from about 5 minutes for a cold machine to 1 hour for a hot machine. Once the shafts are turning the jacking pump can be shut down.

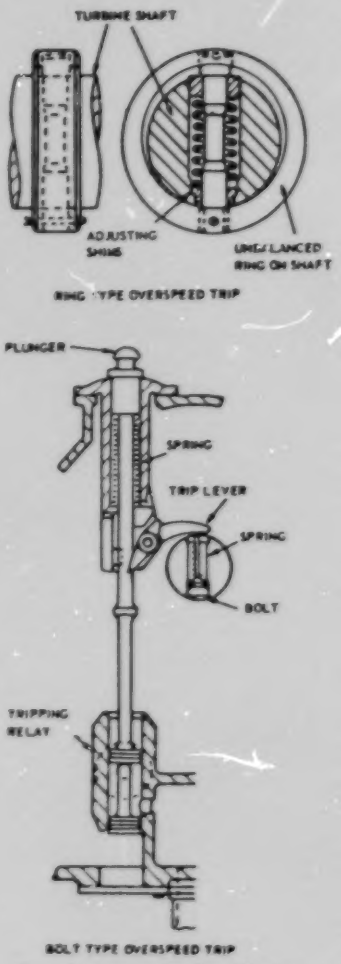
Drains - set all drains on turbine and steam supply line. Then open by-pass steam valves to warm through the steam lines up to the turbine stop valves.

Condensing plant - seal all turbine shaft-glands and build up the condenser vacuum to about 500 to 650 mm mercury using the quick start ejector. Excessive use of gland sealing can produce local over-heating of the turbine shafts and lead to vibration problems. Care should be taken to regulate gland steam to the minimum necessary for complete sealing.

Running Up

Condensate pump - start the condensate or extraction pump and open up the condenser re-circulating valve so that the ejector condensers will have a sufficient supply of condensate as cooling water while the turbine is being loaded.

Emergency trip-gear, Fig. 1 should be tested before admitting steam to the turbine. The turbine stop valves, and their by-passes should be shut, then the emergency stop valves opened fully to their normal operating position and




Emergency Trip Gear
Fig. 1

(PE2-3-6-3)

closed automatically, using the trip-gear. Finally the emergency valves should be set full-open and the turbine run-up commenced.

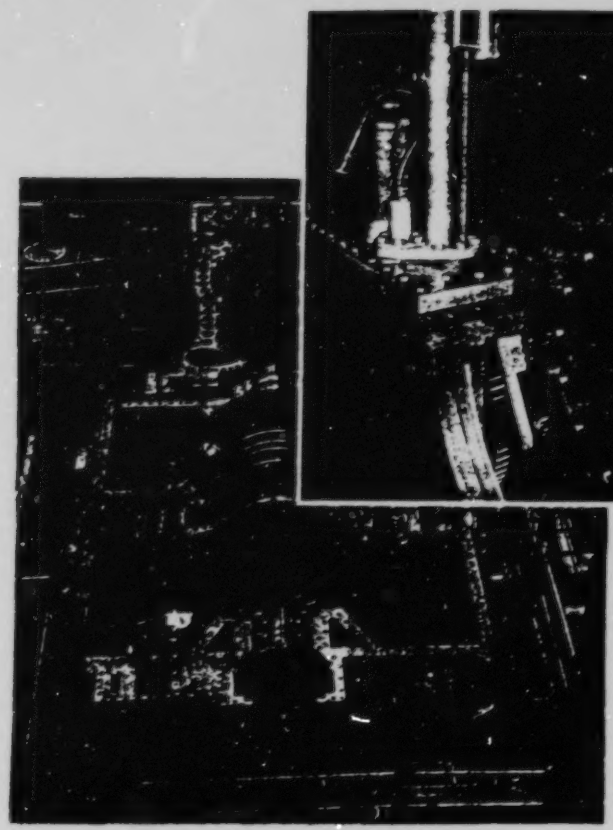
Open the stop valve (or its by-pass) sufficiently to start the turbine rolling, then restrict the steam flow so as to keep the turbine speed in check.

The barring gear, Fig. 2, can be disengaged and shut down. The turbine should be accelerated up to about 300 rev/min in two or three minutes to establish an oil film and held there for a time depending upon its run-up programme.

A machine which does not have a barring gear should be held at 300 rev/min for sufficient time to become warmed through evenly and for any distortion which may have developed after shut down, to be evened out. This may take 15 minutes or longer.

Illustrations of Barring (or Turning) Gear

Fig. 2



Location of underslung barring gear is shown in this view of tandem double-flow turbine being assembled for factory testing. Position of the barring gear at the side of the bearing enables the pinion to engage the shaft below turbine centerline.

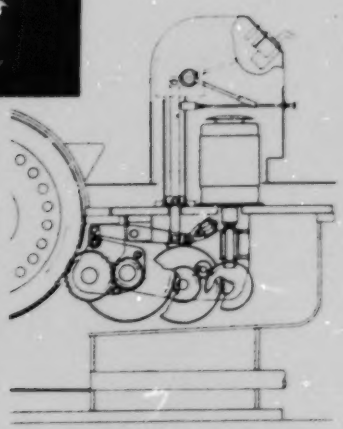


Diagram of side-mounted barring gear and vertical driving motor illustrates their location in relation to the turbine shaft and control console.

(PE2-3-6-4)

Machines which have been turned regularly during their cooling-out period after shut-down, may be run up from rest to about two-thirds of normal full speed without pause at 300 rev/min, taking about 15-20 minutes for the operation. (Note that running at a critical speed will result in machine vibration and therefore these speeds should be passed through without delay.)

During this run up the operator should check that there is no unusual noise or vibration. The main shaft-driven oil pump should come into operation and the auxiliary pump shut down. If the machine is fitted with supervisory instruments these should be watched to ensure that no excessive shaft distortion or displacement is indicated.

A further speed increase up to operating speed should show the governor coming into operation, then, with the machine at full speed and the stop valves fully open it is ready for load.

During the run-up a certain amount of vibration is to be expected at the critical speed or speeds. If this does not smooth out after passing through a critical point, the machine speed should be reduced again until the vibration disappears. If repeated attempts fail to smooth out the vibration the machine may have to be returned to 300 rev/min or even to the barring gear in a further attempt to secure even heating.

Over-speed Trips - when the machine has reached normal running speed and is under the control of its governor the over-speed governor trip should be tested for correct operation. This should be carried out in such a manner that the available steam supply to the turbine is positively controlled at all times, e.g. by a hand-controlled by-pass valve. Steam must not be available to enable the machine to reach a dangerous speed in the event of the failure of any automatic equipment.

The over-speed trip should operate to limit the speed rise to a maximum of 110%. Periodic checks should be made to prove that this equipment operates freely. Opinions of operating engineers as to when is the best time to carry this out follow two schools of thought.

One prefers the test to be carried out when the machine is coming off load and about to be shut down. The idea is that at this time the extra strains imposed by over-speeding are imposed upon a thoroughly warm machine and will therefore be minimized and in case of a failure to operate, maintenance time is then available.

The other, and possibly stronger argument, is that the test should be carried out before a load run, since only through this can the machine be proved safe to operate. Chances of incorrect setting of the equipment during the shut-down are guarded against. A test showing faulty operation of over-speed trips at the time of shutting down would only prove how dangerous the machine had been during the past load run.

Lubricating Oil Coolers - these should be put into service when needed, and the cooling water valves controlled so as to maintain the oil temperature at the maker's specified figure. This will be about 50°C at the turbine bearings. Care should be taken not to overcool the bearing lubricating oil at any time. Cold oil in the bearings can and will cause turbine vibration.

Checks before loading - the machine is now ready for load. The following items should be checked for correct setting and functioning before synchronizing and loading:

Bearing oil pressures and temperatures
Condenser vacuum
Steam drains on turbine and pipework
Condenser (condensate) recirculating valve
Turbine spindle thrust adjustment
All auxiliaries, e.g. extraction pumps, feed pumps, etc.

While loading the machine a careful watch must be kept on bearing temperatures and for signs of vibration, rubbing, unusual noise, or any other such occurrences. The operator's experience and judgment must be relied upon to evaluate such signs of possible trouble as indeed it will be relied upon throughout the turbine operational time.

Many instruments and devices are now available and installed to assist in such evaluations, such as supervisory equipment measuring, shaft vibration, eccentricity or distortion and full use must be made of their indications.

Instructions for Stopping

When decreasing load preparatory to shutting down the machine, the following operations should be carried out.

Set the thrust-adjusting gear to give maximum clearance (where this is fitted).

Open the condenser (condensate) recirculating valves to maintain sufficient flow for ejector condenser cooling.

When all load is off the machine and the main alternator switch has been opened, check the operation of the over-speed trips, if this is required, and close the turbine steam stop valves. The vacuum should be allowed to fall by shutting down the air ejectors or air pump but maintaining a flow of gland-sealing steam until the vacuum has been destroyed. This will prevent the ingress of cold air and oil to the shaft glands, and minimize shaft distortion.

Open all turbine drains.

Check that the auxiliary oil pump cuts in as the turbine speed decreases.

When the shaft has stopped turning engage the barring gear and leave this running for the recommended number of hours as the machine cools down.

In the absence of barring gear, as is usual with smaller machines, the shafts will cool out while standing still. It is particularly important that no steam should be leaking into the cylinders at this time.

Shut the cooling water valves to the oil coolers as soon as possible so as to retain some heat in the oil for the next run-up.

Shut down extraction pumps and circulating water to main condenser.

Normal Operation

Once a turbine has been run up to speed and loaded, the steam temperatures and pressures will remain constant at each stage from inlet through to exhaust. The metal of the rotors and cylinders will approximate towards these temperatures and will become stable in their relative expansions.

The stage pressures and temperatures will be characteristic of that machine at each particular load. They will change with changes in load but otherwise should be constant unless some unusual condition develops. If, for example, some deposit from the steam begins to gather on the blade surfaces, a gradual increase in frictional resistance to steam flow will occur, and this will affect the stage pressure and temperature readings.

It is essential to keep a log of all pertinent temperatures and pressures in order to be able to recognize a diversion from normal figures. A high standard of supervision is required because changes usually take place slowly and are not easily detected.

One of the best means of discovering a trend of change is to set up a basis for comparison of the day-to-day operating figures.

At certain fixed loads, say 25%, 50%, 75% and 100% of full load with inlet and exhaust conditions carefully set, readings should be taken of the steam pressure and temperature at the turbine stop valve, all stage pressures, exhaust back pressure, together with all available readings of spindle locations, vibration from the turbovisory gear, lubricating oil temperature and pressures, etc. These readings should be taken when the machine is in a known state of cleanliness and should be repeated and checked. They can then be used as standards of comparison, and if necessary, printed onto the daily log sheets.

Routine turbine operation on steady load consists of very little more than keeping watch on mechanical conditions such as bearing oil pressures and temperatures and checking for unusual noises or vibrations unless some emergency arises.

A state of emergency can come in many forms to an operating plant. It may not come frequently but when it does the time and the conditions existing always seem to be the worst that could have been chosen. The prime cause of trouble is often camouflaged by its effects. The knowledge and experience of the operating personnel are then called upon to make speedy decisions, an error in which might be extremely costly.

Given steady inlet steam pressure and temperature, the steam conditions through the machine will not vary noticeably. The exhaust vacuum however will be dependent upon the correct operation of the condenser air extraction equipment, the rate of air leakage into the system, the quantity and the temperature of the condenser cooling water.

Exhaust Vacuum

A reduction in vacuum, or conversely stated, an increase in back pressure, adversely affects the economical running of a turbine. Less heat is available per kilogram of steam passing, and so the steam rate, kg per kw-h goes up. As a rough guide it can be said that 6 mm mercury change in back pressure brings a 1% change in the turbine heat rate.

There is an optimum back pressure for each machine which is set during the design stages depending upon the average expected operating conditions, etc. and this is the back pressure at which that machine should run.

Lower back pressures (higher vacuums) than this optimum figure will increase the heat available per kg of steam, but the consequent reduction in condensate temperature will demand more bled steam for feed heating and will result in a net increase in the turbine heat rate (an efficiency reduction).

(PE2-3-6-8)

Higher back pressures than optimum will reduce the heat available per kg of steam. A decrease in bled steam quantity will result from the rise in condensate temperature but will not be sufficient to compensate for the heat loss and the net effect will be a reduction of turbine efficiency.

Accurate measurement of exhaust vacuum is important for the foregoing reasons. The instrument used for this purpose is the Kenotometer. Fig. 3 illustrates the principle and Fig. 4 shows the essential parts of the meter.

The basic objective of the instrument is to record the absolute pressure within the condenser instead of simply the vacuum.

Fig. 3 shows separate vacuum gage and barometer. The barometer indicates the atmospheric pressure in mm of mercury (B). The vacuum gage indicates the vacuum in the condenser, again measured in mm of mercury (M).

The difference between the two columns will give the absolute pressure in the condenser shell.

Stated in terms of pressures,

Absolute Pressure = Atmospheric + Gage.

Thus when the gage pressure is a negative amount (a vacuum) the absolute pressure becomes the difference between the two.

The mercury reservoirs on the vacuum gage and the barometer in Fig. 3 each have the same pressure upon their surfaces so that they could be combined into a single reservoir, or in fact the bottoms of the tubes in both vacuum gage and barometer could be joined to form a single U tube.

This has been done in Fig. 4 and in addition the tube on the vacuum gage has been increased in diameter so that vertical movement in this leg is reduced to a minimum.

Absolute pressure in the condenser is still measured by the difference between the levels in the two legs.

Fig. 4(a) shows the instrument with no vacuum in the condenser. Fig. 4(b) shows the operating condition. The scale is moveable and the zero line must be set to the mercury meniscus in the large diameter leg. The reading h will then give the absolute pressure in the condenser shell in mm of mercury.

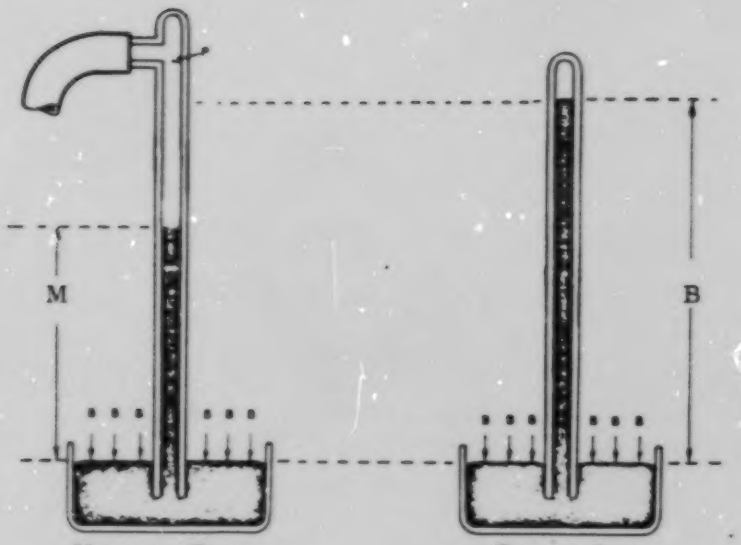
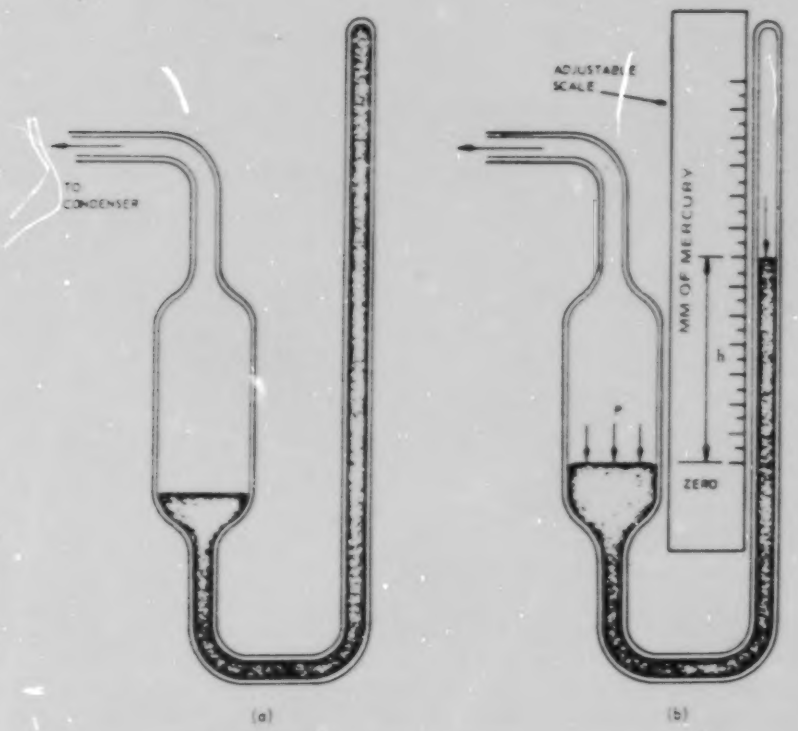



Fig. 3




Kenotometer
Fig. 4

(PE2-3-6-10)

## Blade Fouling

Turbine blading must be maintained in a clean condition if it is to produce the full designed output of the turbine. Deposits which adhere to the blades decrease the turbine efficiency and output and will cause outage or even mechanical damage if not removed.

These deposits develop from carry-over in the steam from the boilers and are principally sodium hydroxide (caustic soda) and silica. Caustic soda melts at 315°C and is soluble in water, hence it will deposit in areas in the turbine where the temperature is below 315°C and where the steam moisture content is insufficient to give a blade-washing effect.

Silica vaporizes at pressures above 4150 kPa and is insoluble in water. Deposits of this chemical may be more generally spread throughout the turbine blading and will also combine with the soluble deposits.

Deposits on turbine blades will gradually reduce the steam passage area and consequently increase the pressure drop through each of the affected stages. Comparison of stage pressure drops with standard figures can be used as an indication of blade fouling.

Removal of these deposits can only be achieved either by washing or by mechanical means. Washing can be carried out without dismantling the machine, mechanical cleaning requires the turbine covers to be lifted and the spindles removed. In either case prevention of carry-over is obviously much more desirable.

Blade washing is done usually on a cold machine and at speeds of rotation not more than 25% of full speed. Moisture-laden steam is introduced through the stop valve with all cylinder drains open and the condensate run to waste. Samples are taken at these points and the procedure continued until these samples show a high degree of purity.

Insoluble deposits will not be removed directly by washing, though they may become cracked and loosened by rapid changes in temperature. Mechanical cleaning methods such as blasting with a mildly abrasive substance will be used for these deposits.

## Blade Erosion

Excessive moisture content in steam flowing through turbine blading can, and will, cause severe erosion. This effect takes place during normal on-load operation and will occur in the low-pressure stages of the turbine.

It is a mechanical effect caused by the impingement of water-droplets upon the leading edges of the turbine blades.

Reference to Lect. 5, Sect. 3, will remind the student of the steam flow through turbine nozzles and blading as represented on vector diagrams.

It will be remembered that the design of a blade is aimed at receiving steam upon the blade surfaces without shock and that in order to achieve this the blade inlet angle was designed to suit the entering steam direction and velocity.

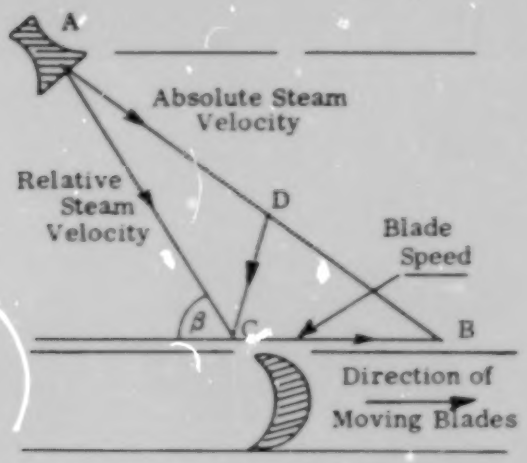



Fig. 5

Fig. 5 shows steam issuing from a nozzle and entering a moving blade ring arranged so that the blade, at its normal running speed, receives steam represented by vector AC. The blade inlet angle is set at $\beta$ degrees so as to match this flow. AC is the resultant of the steam speed AB and the blade speed CB.

A water droplet however, carried in the steam issuing from the nozzle, will flow in the same direction but at a lower speed than the steam because of its inertia.

Let DB represent the water droplet velocity. Then the combination of DB and the blade speed CB gives the speed and direction of the droplet relative to the blade DC.

It will be seen that the blade and droplet will collide at the blade inlet edge and if frequent and heavy collisions occur, erosion will result on this area of the blade. The violence of the collision will depend upon droplet size and upon blade speed.

Experience shows that a moisture content of 10-12% at the turbine exhaust is sufficient to cause erosion.

The problem becomes more acute with increasing machine size. High output machines require large diameter low-pressure wheels to accommodate the vast volumes of exhaust steam. This in turn means that the peripheral speed of the low-pressure blades is high and the impact of water droplets in the steam upon the blades is increased.

(PE2-3-6-12)

Maintenance

A steam turbine will generally not require any more than routine day-to-day maintenance such as care of lubricating oil, inspection of sliding feet etc. for considerable periods of time. Major overhauls requiring removal of the cylinder covers and spindles for examination of blading, etc. are commonly spaced at three to four year intervals. Except when major defects occur, or are suspected, modern high-performance plants should be kept in service for as long as economically justifiable. In other words, the plant should be kept in service until its performance deteriorates to a point where the cost of outage for overhaul is out-weighed by the cost of continuing to run the plant at a low efficiency.

The major parts of a turbine requiring maintenance under overhaul are as follows:

Blading

The blading may have been washed while in service following steam consumption checks, or stage pressure changes which indicated fouling, but it should be prepared for complete cleaning during the overhaul. The cylinder covers will be taken off and inverted, the turbine spindles will be lifted out, and, where necessary the top and bottom half of diaphragms will be removed from the cylinders.

Blading should be inspected for evidence of corrosion, erosion and mechanical rubbing. Blades will be "dressed" as necessary, and badly damaged sections will be replaced.

In low-pressure stages lacing wires must be rebrazed or replaced as necessary. Shroud bands in high-pressure and intermediate-pressure stages should be inspected for signs of rubbing and dressed up or replaced.

All blades should be inspected for cracks in the blade or at the root, particularly in the low-pressure stages, using one of the proven crack-detection methods.

Glands

During operation any deterioration in the condition of the shaft glands will have been indicated by an increase in the amount of steam required for sealing. During overhaul, gland packings must be cleaned, straightened where necessary, and adjustments made to restore correct clearances. Spring-loaded sections are usually set up and dressed to fit correctly on a mandrel representing the turbine shaft. Gland steam supply pipes, vent pipes and drainage holes should be examined for cleanliness.

(PE2-3-6-13)

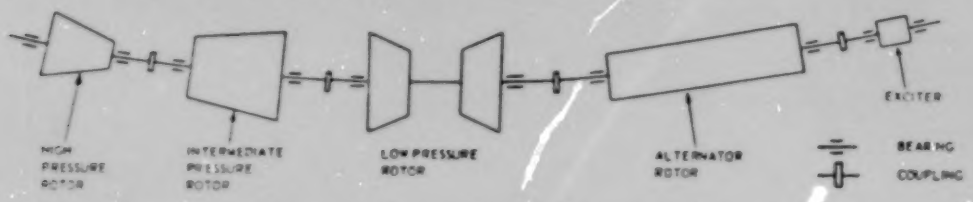
Diaphragms

Diaphragms should be inspected for cracks and checked for distortion, erosion or rubbing. The casing groove landings should be checked to ensure that the diaphragms fit properly. The nozzles should be cleared of any deposits and the edges dressed.

Alignment

Because of the speed of the rotating masses and the large out-of-balance forces which can appear as vibration, the alignment of a large modern turbine is very carefully carried out when erected.

The general principle of alignment is that, assuming the coupling faces to be true with the shafts, the shafts are aligned in such a way that a continuous curve is formed, with their natural deflections, from governor to exciter. This point is illustrated exaggerated in Fig. 6. The shafts will retain their natural deflection at any speed other than the critical speeds. Adjustment to give correct alignment is carried out by the adjustment of bearing positions to match this static deflection of the shaft.




Alignment Curve for Turbo-generator
Fig. 6

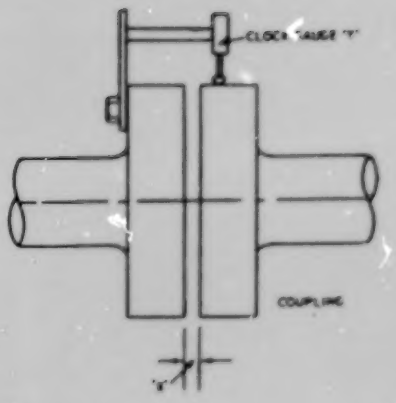
It is not necessary to know the shaft deflection curve. Correct alignment is obtained by accurate measurements between coupling faces and over the coupling periphery. When equal measurements are obtained by clock gage or feelers at four points $90^{o}$ apart round the coupling periphery at locations x and y, Fig. 7, then correct alignment can be assumed, provided that the coupling faces and periphery are 'true' with the shaft.

Two general cases of misalignment occur:

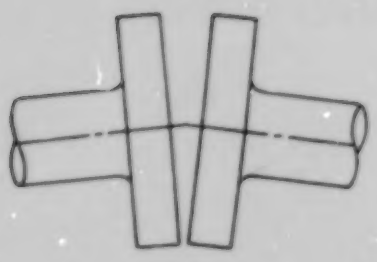
(1) The axes of the two shafts may meet but may not be in a straight line, as shown in Fig. 8.

(2) The axes may be parallel but may not be in line, see Fig. 9.

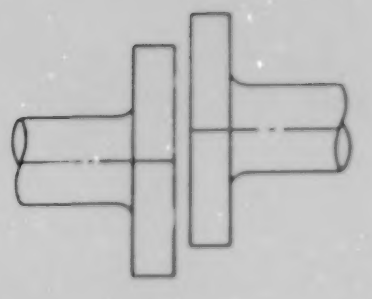
Most manufacturers supply an alignment gage for a particular machine which consists of a plate with a gap to cross the coupling and which has two true edges accurately aligned as illustrated in Fig. 10. When applied across a coupling, if both edges are wholly in contact with the shafts on each side, the correct alignment is established. Misalignment of the type indicated in (1) and (2) above will be revealed as in Fig. 10. The gage can also be laid on the horizontal joint to check horizontal alignment.



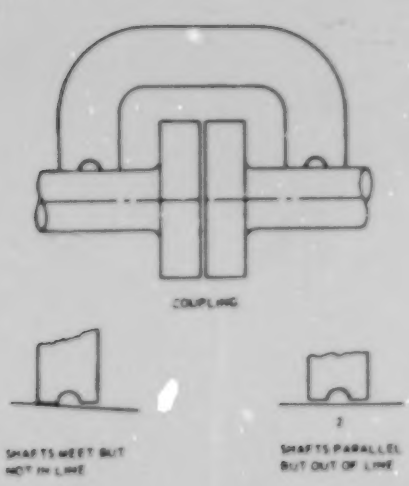

Alignment Measurements at Coupling
Fig. 7




Shafts meeting but Out-of-line
Fig. 8




Shafts parallel but Out-of-line
Fig. 9




Use of Alignment Gage
Fig. 10

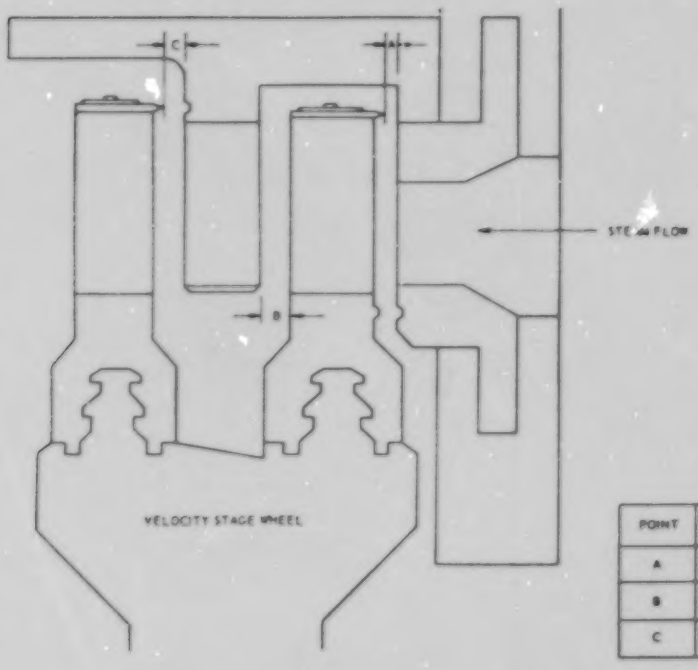
(PE2-3-6-15)

## Clearances

The efficient operation of a turbine depends to a large extent on the maintenance of the correct clearances between fixed and moving elements. Excessive clearances result in increased steam consumption while reduced clearances may result in blade rubbing.

When a turbine is erected the clearances are carefully set and a record is kept at the plant. When the top halves of the casing are removed the clearances should be checked against the record. Care must be taken to ensure that the rotors are in the running position when taking measurements. Provision is usually made to move the rotor axially to a position for lifting from and returning to the casing.

Particular care is necessary with the clearances at the velocity stages which are frequently fitted to the high-pressure end of impulse machines, as shown in Fig. 11. A thorough check of clearances is essential if any replacement blades, nozzles or packing rings have been fitted.




| POINT | CLEARANCE mm |
|---|---|
| A | 2 |
| B | 5 |
| C | 2 |

Velocity Stage Clearances
Fig. 11

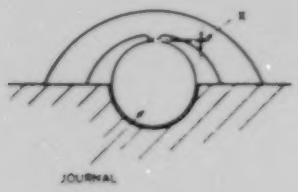
(PE2-3-6-16)

## Bearings

A thorough examination is made of bearings for wear, grooving of the bearing metal and shaft, loose bearing metal, correct contact surface and possible evidence of electrolysis. Modern bearings are of the spherically-seated type and their fit in the housing should be checked for tightness and alignment and adjustments made if necessary.

The condition of oil orifices, including the area of high-pressure jacking oil, oil throwers, baffles and the cleanliness of all oil and water passages should be checked. It is usual to measure and record bearing clearances. For this purpose a bridge gage is used as shown in Fig. 12 and the measurement at X is compared with the records.

Variations will indicate bearing wear or settlement. A typical permissible clearance is 0.025 to 0.05 mm per 25 mm diameter of journal.



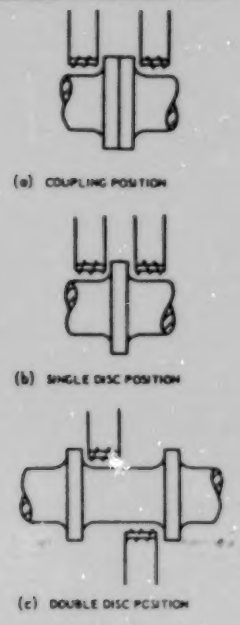

Use of Bridge Gage
Fig. 12

## Instruments and Controls

Mention was made earlier in this lecture of exhaust vacuum measurements because of its importance in turbine operation; the other instruments used for this purpose can be grouped under the general heading "Turbine Supervisory Equipment". This consists of instruments to indicate and record differential expansions between rotors and casings, rotor eccentricity, steam to metal differential temperatures, bearing vibration, speed and load.

## TURBINE SUPERVISORY EQUIPMENT

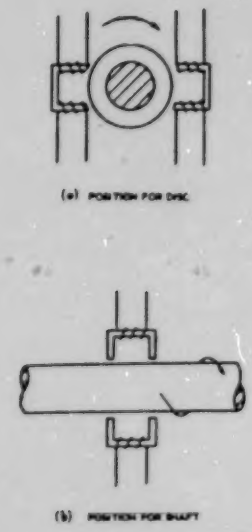
### Arrangement of Detectors




Arrangements of Axial Expansion Detector
Fig. 13

The indicators and recorders of differential expansion and eccentricity are actuated by electronic circuitry from specially designed inductive detectors placed near appropriate moving parts of the rotating shaft.

Fig. 13 shows three alternative positions for the axial or differential expansion detectors, while Fig. 14 shows two alternative positions for the eccentricity detectors. Other methods of locating these detectors can be used provided that the air-gap variation between the coupling, collar or shaft, and the detector is definitely caused by the distortion which is to be measured.

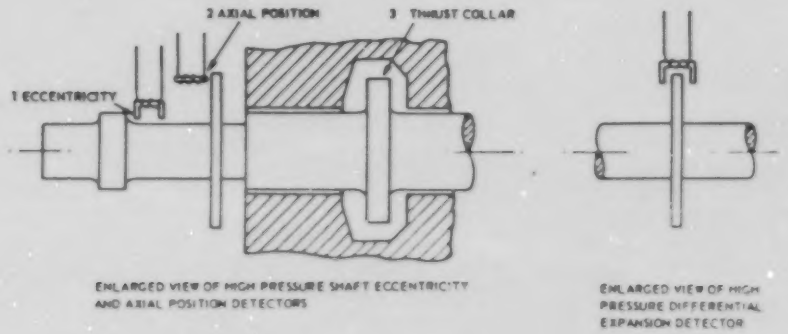


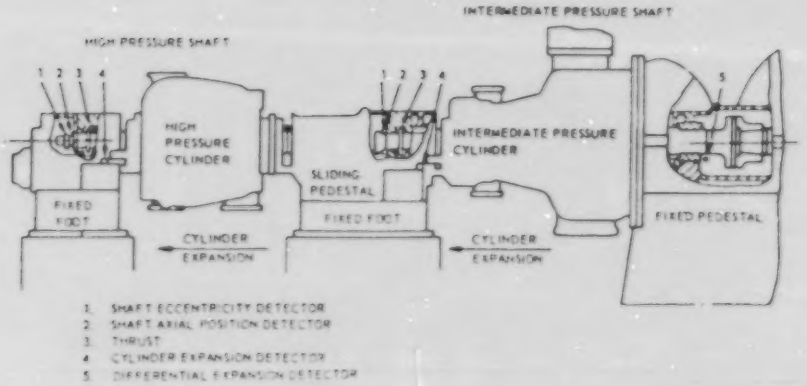

Arrangements of Eccentricity Detectors
Fig. 14

(PE2-3-6-18)

The relative positions of the detectors on the high-pressure or intermediate-pressure cylinders of a large turbine are shown in Fig. 15. Cylinder movement detectors are also provided in this particular arrangement.

Most detectors are subjected to considerable heating because of the high temperature attained by the cylinder casing and bearing housings. Since it is essential that these detectors should be reliable in operation, care is taken in construction so that they withstand heat. The connecting leads are usually insulated with glass fabric and protected by metallic tubing.


2 AXIAL POSITION
3 THRUST COLLAR
1 ECCENTRICITY
ENLARGED VIEW OF HIGH PRESSURE SHAFT ECCENTRICITY AND AXIAL POSITION DETECTORS
ENLARGED VIEW OF HIGH PRESSURE DIFFERENTIAL EXPANSION DETECTOR




Relative positions of Detectors on Turbine
Fig. 15

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Description of the Supervisory Equipment

The essential items are as follows:

(1) Indication and recording of vertical eccentricity.
(2) Indication and recording of horizontal eccentricity.
(3) Indication and recording of differential expansion.
(4) Indication and recording of shaft speed.

Other refinements now being added in some equipment are:

(5) Indication of mean shaft position in bearing with respect to oil film.
(6) Indication of electrical output.
(7) Indication of turbine casing temperatures.
(8) Indication of vibration amplitudes at bearings.

Here, items (1) to (4) are discussed and the method of obtaining readings is described.

The detectors are iron-cored inductances arranged in matched pairs so that the air-gaps with respect to the rotating disc, or other steel part of the shaft, are equal under correct conditions. When the shaft becomes distorted, these air-gaps are no longer equal and the value of the inductance of the detectors is changed.

The detectors form two arms of a Wheatstone bridge, the other two arms being resistances.

The speed indicator is usually supplied from a tacho-generator direct-coupled to the shaft. The a-c output of this small generator, which has a permanent magnet field, is linear with speed and has sufficient power to operate the indicating and recording instruments.

Use of Supervisory Equipment

If supervisory equipment is not available during initial stages of starting-up, differential expansion can take place, as for example, in Fig. 16. The introduction of gland steam and flange steam alternatively will help to keep the turbine in a smooth running state, depending on the skill and experience of the turbine operator. Eccentricity may, however, ultimately develop if the temperature gradients become excessive, because of these "swings" in expansion between wheel and diaphragm. When supervisory equipment is available to guide the turbine operator, the machine can be run up with the "swings" of differential expansion very much reduced as is shown in Fig. 17, and it should be possible to keep eccentricity below 0.125 mm.

Vibration

It is sometimes possible for a turbine shaft to run with appreciable eccentricity all unknown to the operator. Vibration measurements, while giving a good guide to the general running of a turbine, may not give a true interpretation of the rotor condition. Cases have been known when sets have run without causing undue vibration although several blades were missing from the wheels. An acceptable condition for reasonably good operation is an amplitude of between 0.025 to 0.05 mm, but rough running is soon reached above the latter figure.

(PE2-3-6-20)

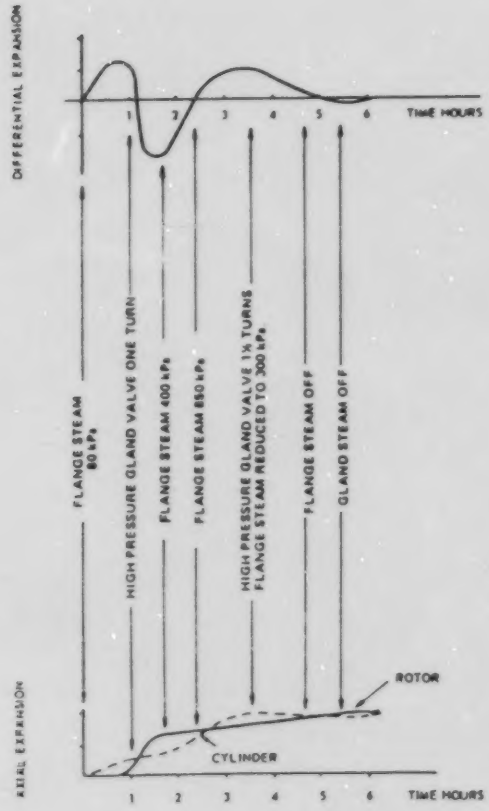
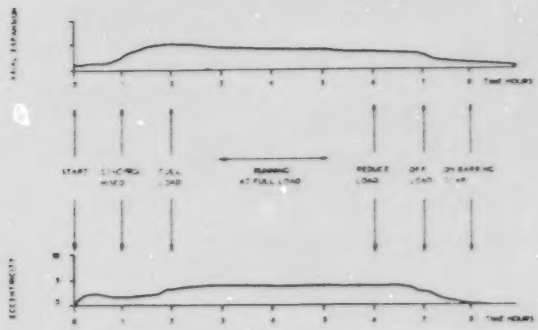



Fig. 16

Differential and Axial Expansion during a Turbine Run-up without Supervisory Equipment



Axial Expansion and Eccentricity during a Start-up loading and Shut-down of a large Turbine using Supervisory Equipment Fig. 17 (PE2-3-6-21)

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## Turbine Controls

In the previous pages, the operation and use of the kind of instruments associated with measurements required on a turbo-generator, has been described. As an illustration of the uses of the various instruments, a typical turbine instrument scheme is described below.

### Typical Turbine Control Panel
for 60 MW set, 6200 kPa gage, 480°C type

(1) Output

An induction pattern wattmeter is connected in series with one of the tariff meters to indicate instantaneous load, MW.

(2) Pressures

Pressure gages are used to show:

- Steam pressure before and after throttle
- Steam pressure after first wheel (Curtis stage)
- Gland steam leak-off
- All bled steam pressures (above atmospheric)
- Pressure to deaerator
- High-pressure turbine exhaust
- Bearing oil pressure
- Relay oil pressure for operation of governing mechanism.

(3) Vacuum and Low Pressure

Kenotometer instrument for condenser absolute pressure. Sub-atmospheric pressures are given by diaphragm gages.

(4) Levels and Positions

The levels of water (condensate) in the reserve tank and deaerator vessel, are transmitted electrically to deflectional instruments. Positions of circulating water valves are sometimes shown on indicating instruments on the turbine panel since it is necessary to adjust them to obtain optimum vacuum, depending on circulating water temperature.

(5) Temperatures

Turbine Stop Valve (T.S.V.) steam and the final feedwater usually have separate large dial instruments. Recording instruments are sometimes used for T.S.V. temperature. Other temperatures on the turbo-generator set are given on two multipoint instruments.

(6) Condensate Purity

It is usual to provide an indicating instrument supplied from the "Dionic" meter. Dissolved oxygen meters and hydrogen ion concentration meters are usually mounted separately elsewhere.

(7) Supervisory Panel

This is sometimes incorporated into the turbine control panel; sometimes it is a separate unit. Its instruments were described in previous pages.

(8) Indicators and Alarms

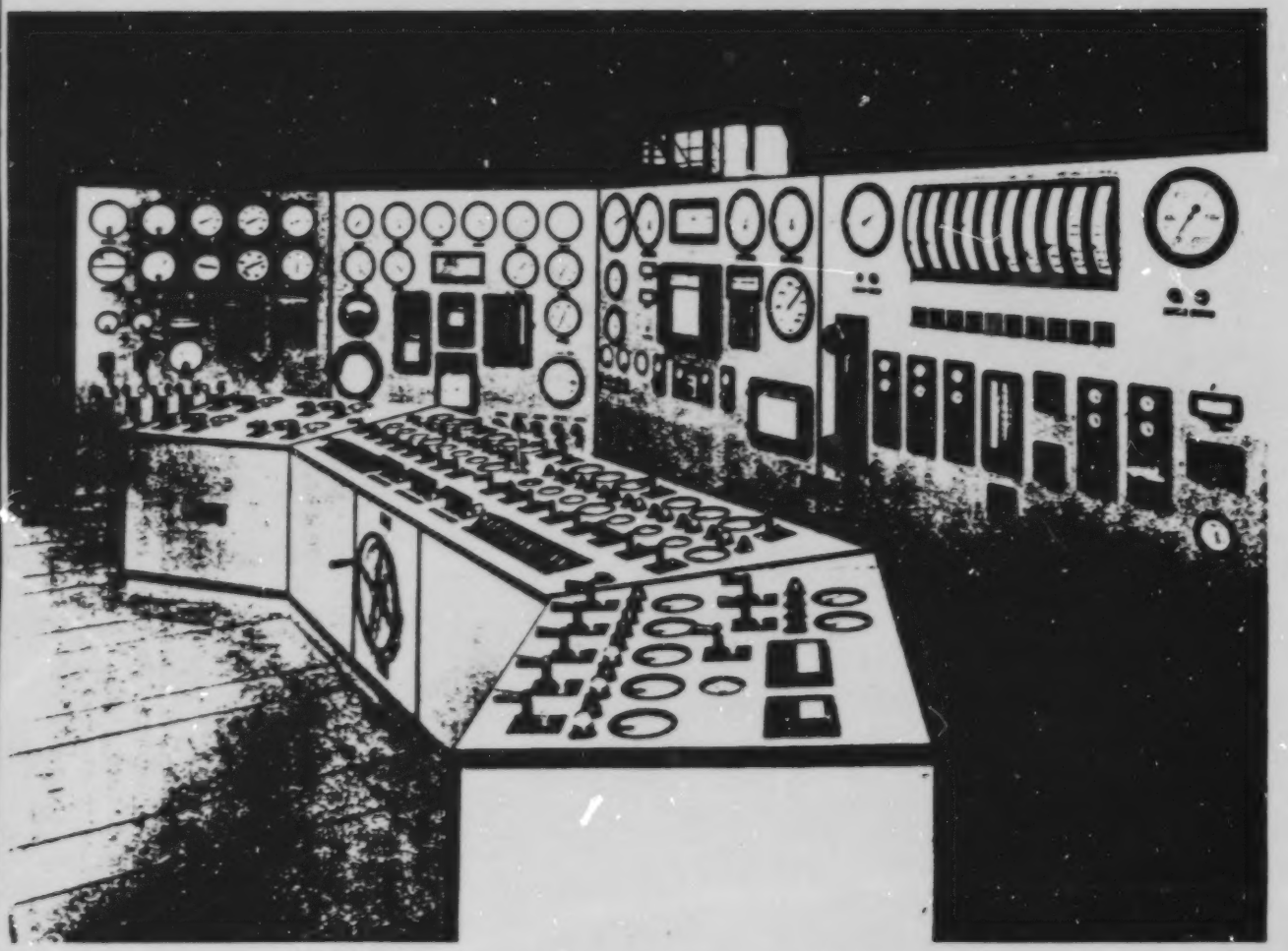
While these are not classified as instruments, it should be mentioned that the malfunctioning of parts of the plant is usually shown up on illuminated indicator panels, for example, excessive exhaust temperature, high bearing temperature, low oil level in reservoir, high water in high-pressure heater, high and low water in storage vessels, high-water level in condenser, high conductivity in condensate, low-hydrogen purity in generator and loss of seal oil supply.

Fig. 18 is a photograph which illustrates a typical 6200 kPa gage, 480°C combined boiler-turbo-generator gage board. This layout is efficient and compact.

Turbine panels are on the left-hand side of the operator. Pressure gages and position indicators are at the top. In the middle, from the left to right, may be seen the two multipoint temperature instruments side-by-side, the water purity meter, eccentricity records, and the Kenotometer, with vertical scale on its right-hand side.

On the extreme left of the panel are operating handles for motor-operated valves and pumps. On the console itself are further operating mechanisms for controlling auxiliary plant. Governor and generator voltage are controlled from a central control room.

(PE2-3-6-23)



Combined Boiler and Turbine Unit Control Panel
Fig. 18

## Turbine Automatic Control Equipment

The turbine instrumentation discussed in earlier pages indicated and recorded the variations in certain temperatures, pressures, bearing vibrations, etc.

If now mechanical drives were fitted to the steam valves, drains, etc. normally manually-operated during a turbine run-up, and the turbine supervisory instruments arranged to monitor conditions, a program could be set up to automatically start and run up a turbo-alternator set.

The following pages will describe such a system of turbine control developed by Associated Electrical Industries Ltd. (AEI), and given the name ACTRUS from the description, Automatically Controlled Turbine Run-up System.

## Turbine Run-ups

Turbine run-ups are divided broadly into two types, cold and hot. Cold starts occur after prolonged periods of shut-down, such as annual overhauls. Hot starts usually apply to the run-ups following an all-night or week-end shut-down, when the turbine metal will have retained a large part of its heat content.

The main effect of this retained heat on starting, is that the differential temperature between the turbine metal and the incoming steam is reduced, thereby permitting the use of higher steam temperatures and pressures with a resulting quicker run-up. Programmes as short as a few minutes are envisaged for hot starts, while those for cold starts, on the other hand, may take up to several hours.

For cold starts the usual form of programme is:

(1) A run-up from barring speed - about 40 rev/min - to 1200 rev/min at an intermediate rate of acceleration.

(2) A "soaking" period in which the speed increases slowly from 1200 rev/min to 1700 rev/min.

(3) A final run-up to 3600 rev/min at a high rate of acceleration.

The purpose of the soaking period is to warm up the turbine metal and reduce differential temperature to a minimum before proceeding to full speed. For reasons already mentioned, the soaking period can be omitted for a hot start and this, together with a faster rate in the early part of the run-up, results in a considerably shorter programme.

Under normal conditions the pre-set programme is followed without interruption. But should abnormal conditions arise, the programme may have to be interrupted or even reversed. This may be done either manually or by means of the automatic supervisory equipment built into ACTRUS and operating on signals derived, for example, from turbine vibration, eccentricity of the rotor, or differential temperatures. Should any one of the supervisory signals exceed a certain limit the programme is interrupted and the turbine speed is held steady. This situation is known as 'HOLD'.

If the supervisory signal continues to rise to a second limit, the programme is reversed and the by-pass valves are made to close by reversing the motor operating gear. This second situation is known as 'RUN-DOWN'.

In cases of greater emergency (i.e. over-riding signals from eccentricity or vibration detectors) the by-pass (and governor) valves are tripped by releasing relay oil to drain. A more detailed description of these conditions, and certain limitations thereto, is given in a later section headed 'Automatic Supervisory Equipment'.

Fig. 21 shows the equipment diagrammatically. Note that it is divided into three sections: Speed Control, Programme, and Supervisory Equipment.

Briefly the Programme unit sets the rate of increase of turbine speed during the run-up. The Speed control opens the steam valves. The Automatic Supervisory equipment monitors the conditions obtaining in the turbine and will interrupt the programme in the event of abnormal conditions, such as excessive vibration, differential expansion, etc. The following pages describe each section in turn.

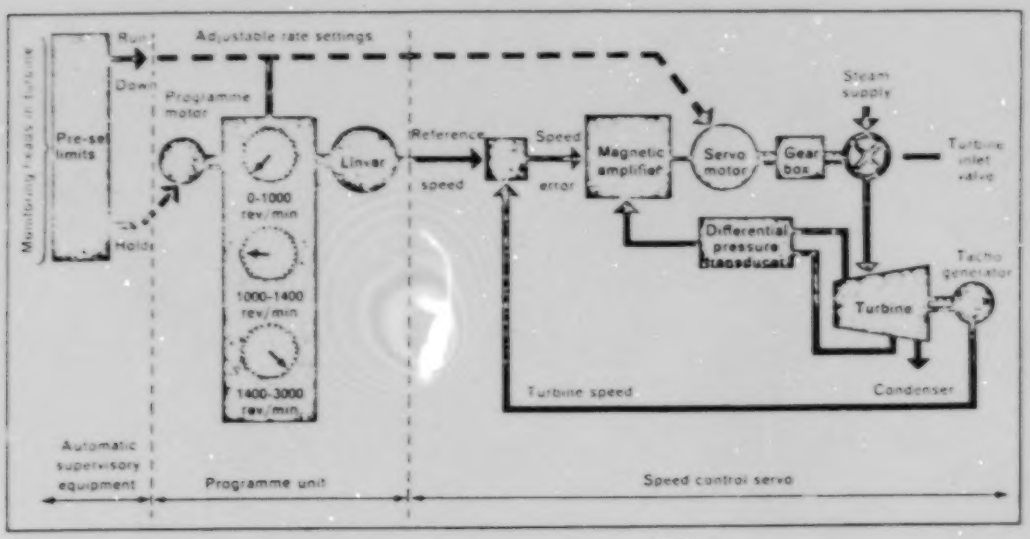



Fig. 21

(PE2-3-6-26)

Programme Unit

The object of the programme unit is to produce a reference voltage which, during run-up increases at pre-set rates from zero to a value representing full turbine speed. This reference voltage is derived from a "Linvar" rotary inductive device giving an output proportional to its angle of rotation over the range 0-80°.

The rotor of the Linvar is driven by one of three constant speeds motors, each having an adjustable gear ratio. A cam, geared to the linvar drive shaft, selects the three speed ratios in turn, the change-overs being made at 1200 rev/min and 1700 rev/min. The Linvar is therefore driven at the first pre-set speed for turbine speeds up to 120 rev/min, at the second speed for turbine speeds from 1200 - 1700 rev/min and at the third speed for turbine speeds from 1700 - 3600 rev/min.

The pre-set speeds are manually-set before start-up depending on turbine conditions - in particular the metal temperature of the high-pressure cylinder. When the metal temperature is high, as it would be after an overnight shut-down, the slow soaking period provided by the second speed ratio is not required and the programme unit can be set to change over directly from the first ratio to the third at a turbine speed of 1200 rev/min. In this way the voltage from the Linvar rises from zero to a maximum and the speed control servo causes the turbine speed to increase in a similar way.

Fig. 22 shows an illustration of the unit.

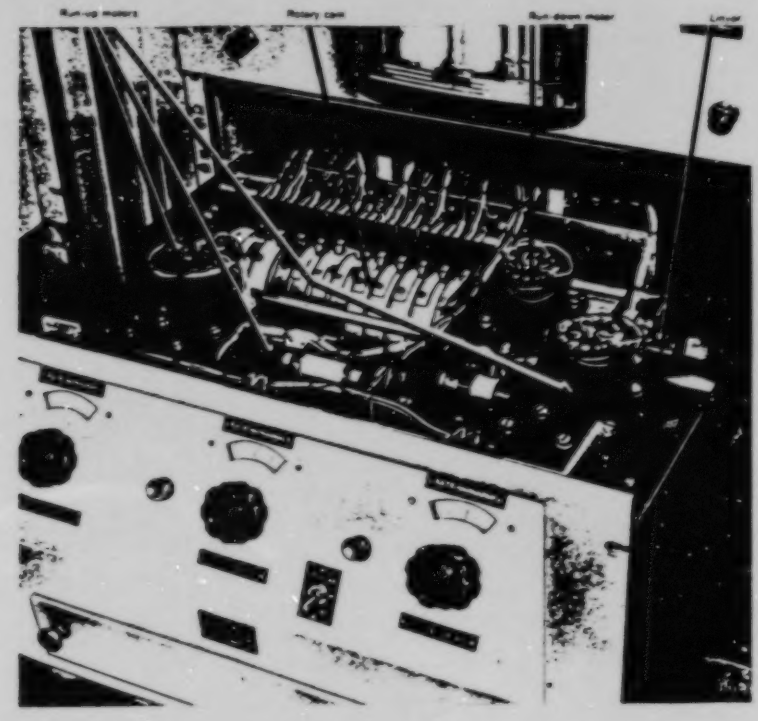


Fig. 22

(PE2-3-6-27)

Speed Control Servo

Referring to Fig. 23, the turbine speed is caused to follow the desired programme by servo control of the combined stop and emergency by-pass valves. An a-c tach-generator, driven by the turbine, produces a voltage output proportional to turbine speed. The difference between this voltage and the reference voltage (known as the speed error) passes through a magnetic amplifier and is then used to control the speed and direction of the servo motors.

If the turbine speed is too high the valves are driven in a closing direction to reduce the flow of steam into the turbine; when the turbine speed is too low the valve openings are increased to produce acceleration. This control, therefore, constitutes a closed-loop or servo system, turbine speed being controlled by valve opening which is itself controlled by turbine speed-acting so that the speed error always tends to zero.

In order to ensure that the rate of opening of the two by-pass valves is equal, their positions are measured by synchro elements and a balancing differential signal is fed back to the amplifiers in the servo control unit. Where the steam connections to the combined stop and emergency by-pass valves are commoned on both sides of the valves it is possible to run-up by opening one valve only. In this case it is usual for both valves to be fitted with servo equipment, but ACTRUS controls only one of them during the course of run-up. Equal wear of the valves can be achieved by dividing the run-up between them during the lifetime of the set.

Due to the inertia of the turbine and alternator rotors, the speed will change rather slowly and the control described would be relatively underdamped. To overcome this an additional voltage proportional to the pressure drop through the turbine is generated in a pressure transducer and fed back into the magnetic amplifier. Due to rapid response of pressure to valve opening, this feed-back produces a well damped control.

If conditions in the turbine required that a 'HOLD' should be made in the programme, the drive to the 'Linvar' is interrupted. The speed reference is thus held constant and consequently no increase in turbine speed is permitted until the programme is restarted.

Where conditions in the turbine are more critical, a 'RUN-DOWN' may be necessary. In this case the programme is interrupted as before, but here two additional actions occur. In the first place the speed control servo receives a bias signal which causes the steam valves to close at the full rate of the servo motors. The set then slows down at a rate dependent upon its own inertia and the friction forces operating. As this takes place an additional motor in the programme unit is energised and controlled by the speed error signal in such a manner that the reference signal is at all times proportional to the turbine speed. Thus, when run-up is re-initiated, the turbine and reference speeds are already equal and normal run-up can proceed.

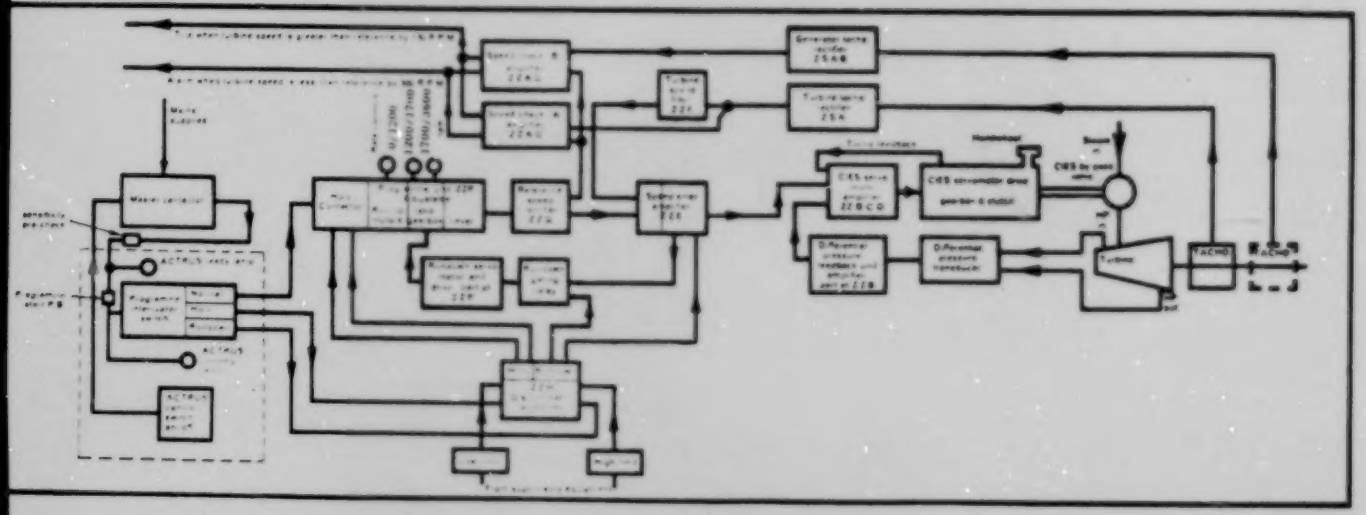


Fig. 23

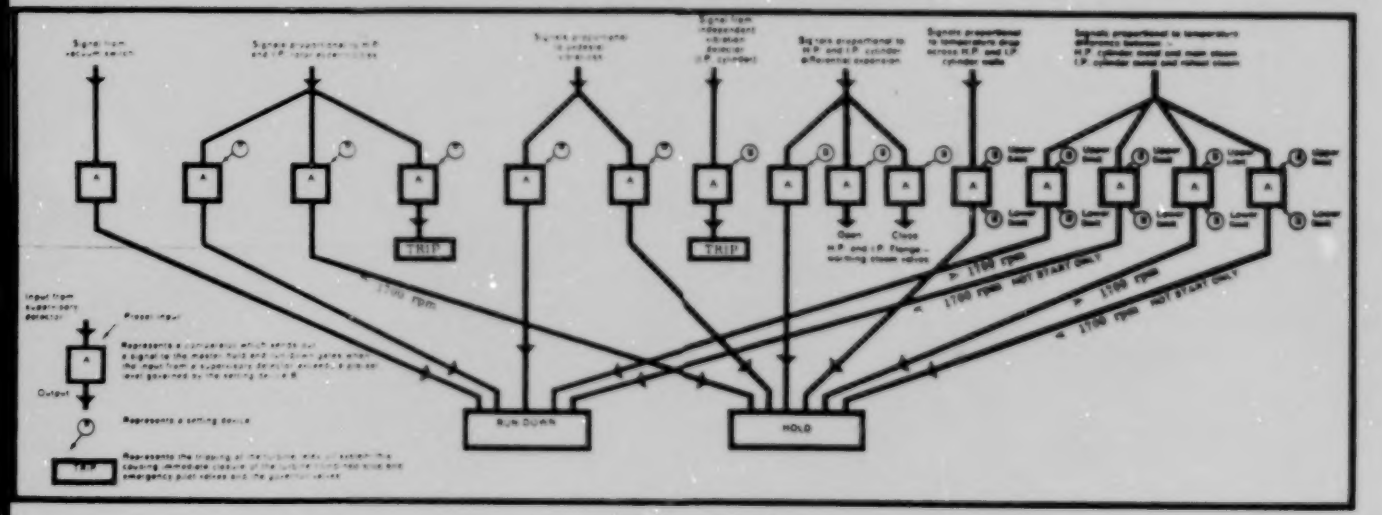



Fig. 24

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Automatic Supervisory Equipment

Detectors fitted on the turbine ensure constant supervision of vibration, rotor eccentricity, differential temperatures, condenser vacuum, etc. The outputs from these devices are fed to the turbovisory equipment mounted in a separate cubicle alongside ACTRUS. This operates various recorders and indicators whenever the turbine is running. During run-up the turbovisory signals are also fed into comparators (A), Fig. 24 on the previous page, where they are compared with references which may be pre-set by means of setting devices (B). These comparators are actually sensitive moving-coil relays with adjustable contacts for use as reference values, which are finally fixed during commissioning to suit the particular set concerned.

When the incoming signal exceeds the reference (that is, when the limits are exceeded), the comparators initiate self-sealing relays which pass on the signal to the master hold or run-down gate. This, in turn, sends the appropriate control signal to the speed control unit and, in the case of differential expansion, it also initiates operation of the H-p and I-p flange-warming valves. As the self-sealing relays will only reset when the supervisory signal has dropped by about 10%, the possibility of indefinite action around the set point is eliminated.

To give a typical example: if pedestal vibration exceeds a pre-set amplitude, a HOLD is applied to the programme by stopping the programme drive motor. When the vibration again comes within limits, the programme is resumed. If the vibration should exceed a second limit, the steam valve is closed causing the turbine speed to fall. At the same time a run-down motor in the programme unit drives the Linvar so that its output is always matched to that of the turbine tachometer. Conditions are then correct for resuming the run-up programme when the vibration has dropped to an acceptable value.

QUESTION SHEET

POWER ENGINEERING

Second Class
Sect. 3, Lect. 6

1. Comment upon the precautions which should be taken before starting a new steam turbine for the first time after completion of erection.

2. Briefly describe the steps to be taken when running up a central station turbo-alternator.

3. What faults can cause loss of vacuum in a steam turbine condenser? How would you discover and correct these?

4. Sketch and describe an instrument used for measuring the absolute pressure in a turbine condenser.

5. Why measure absolute pressure rather than vacuum, in a turbine condenser?

6. What is meant by blade fouling? How can this be prevented?

7. Distinguish between blade fouling and blade erosion. What precautions can be taken to avoid the latter?

8. What is Turbine Supervisory equipment? What measurements does it make?

9. Explain how a turbine shaft vibration and expansion can be measured and recorded and how this information is put to use during the turbine operation.

10. Give a brief outline of the operation of a fully-automated control system used for running a turbo-alternator up to speed, ready for loading.

(PE2-3-6-Q)

INSTRUCTIONAL LEARNING SYSTEMS

# 8.5

GAS TURBINES

## Goal:

The apprentice will be able to describe gas turbines and their operationl

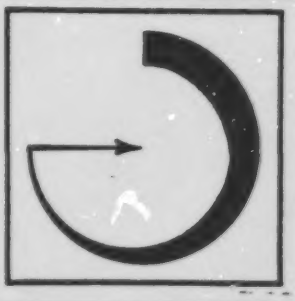
## Performance Indicators:

1. Describe types of gas turbine systems.
2. Describe functions of regenerators, intercoolers and reheaters.
3. Describe components of a gas turbine system.

# Study Guide

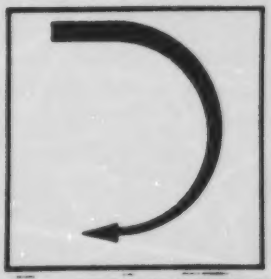
* Read the goal and performance indicators to find what is to be learned from package.
* Read the vocabulary list to find new words that will be used in package.
* Read the introduction and information sheets.
* Complete the job sheet.
* Complete self-assessment.
* Complete post-assessment.

# Vocabulary



* Axial flow type
* Centrifugal type
* Closed cycle systems
* Combustion chamber
* Dual shaft machines
* Dynamic type
* Ignition rod
* Inner jacket
* Intercoolers
* Open cycle systems
* Outer jacket
* Positive displacement compressor
* Regeneration
* Reheaters
* Swirler
* Turbine

# Introduction



Power plants are sometimes located in areas where steam turbines are not feasible or the purpose of the plant does not justify a steam powered operation. Gas driven turbines are often more suitable in those cases.

A gas turbine operation has certain characteristics that make them suitable in some cases. They offer a simple plant layout with low installation costs. The gas turbine requires a lighter foundation than a steam plant and a much smaller location area. These characteristics make them suitable for electricity generation up to 30 megawatts of output. Gas turbines find wide usage in aircraft, oil pipeline stations and operation of ships and railroads.

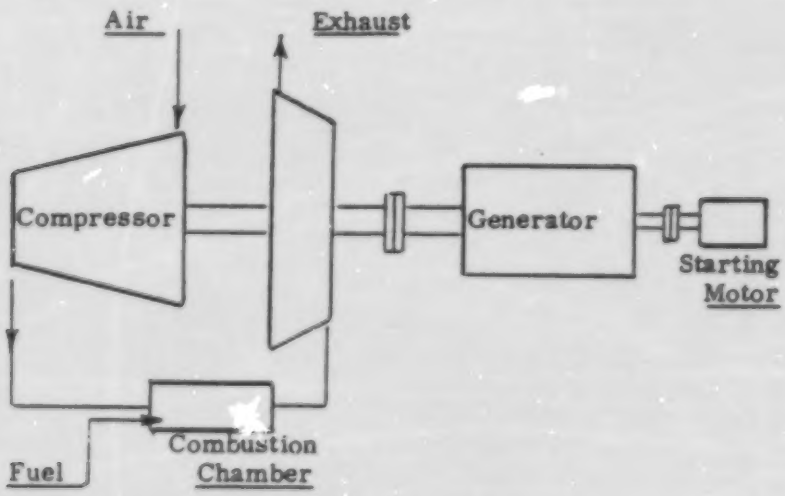
# Information

The gas turbine operates by passing compressed, hot air through the blades of a turbine. The air compressor requires about 2/3 of the energy produced by the turbine. The compressed air enters the turbine at some 700 C. Higher temperatures increase the efficiency of the turbine but excessive heat will damage the blades.

## TYPES OF GAS TURBINES

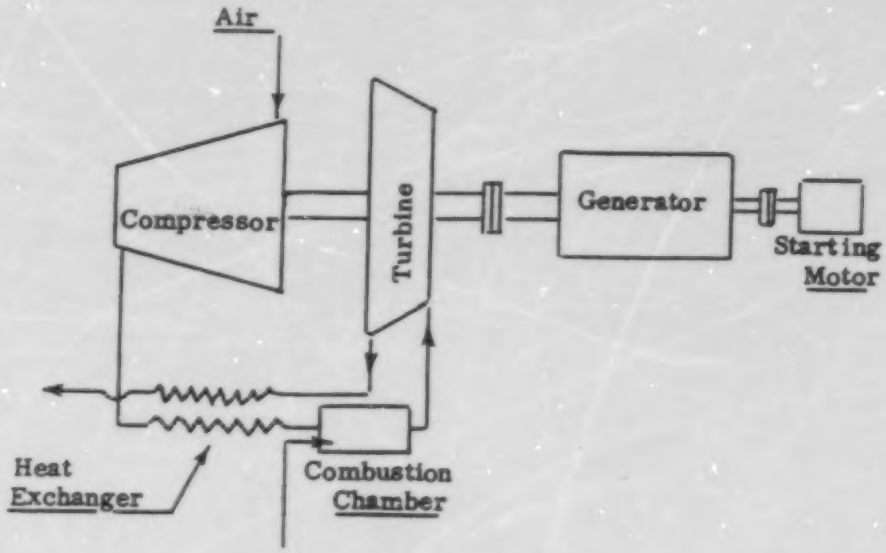
### Open Cycle Systems

A gas turbine plant is composed of a compressor, combustion chamber and a turbine. Open cycle systems pull inlet air from the atmosphere and dump exhaust air back into the atmosphere. Some units have heat exchangers that save the exhaust heat and use it at the inlet. The process of saving exhaust heat is termed regeneration. A simple open cycle gas turbine plant is shown in the following diagram.



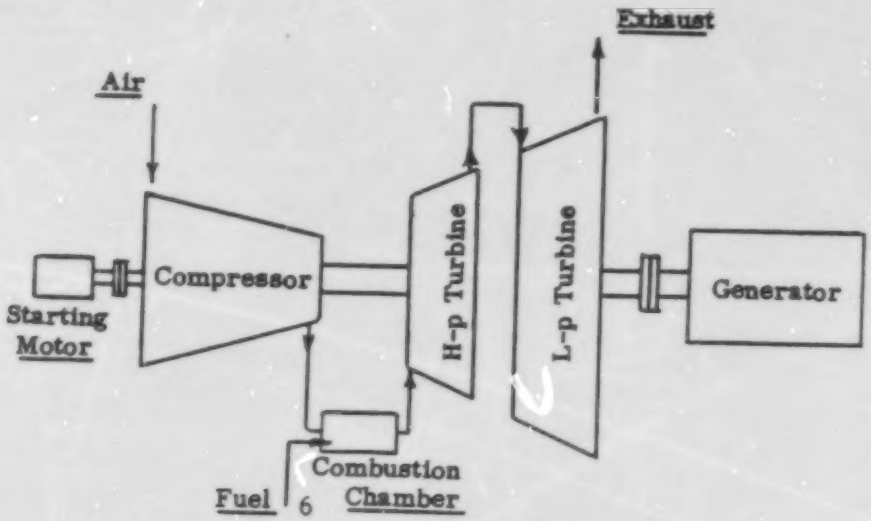
# Information

A simple unit with a heat exchanger added is shown below.




Dual Shaft Machines

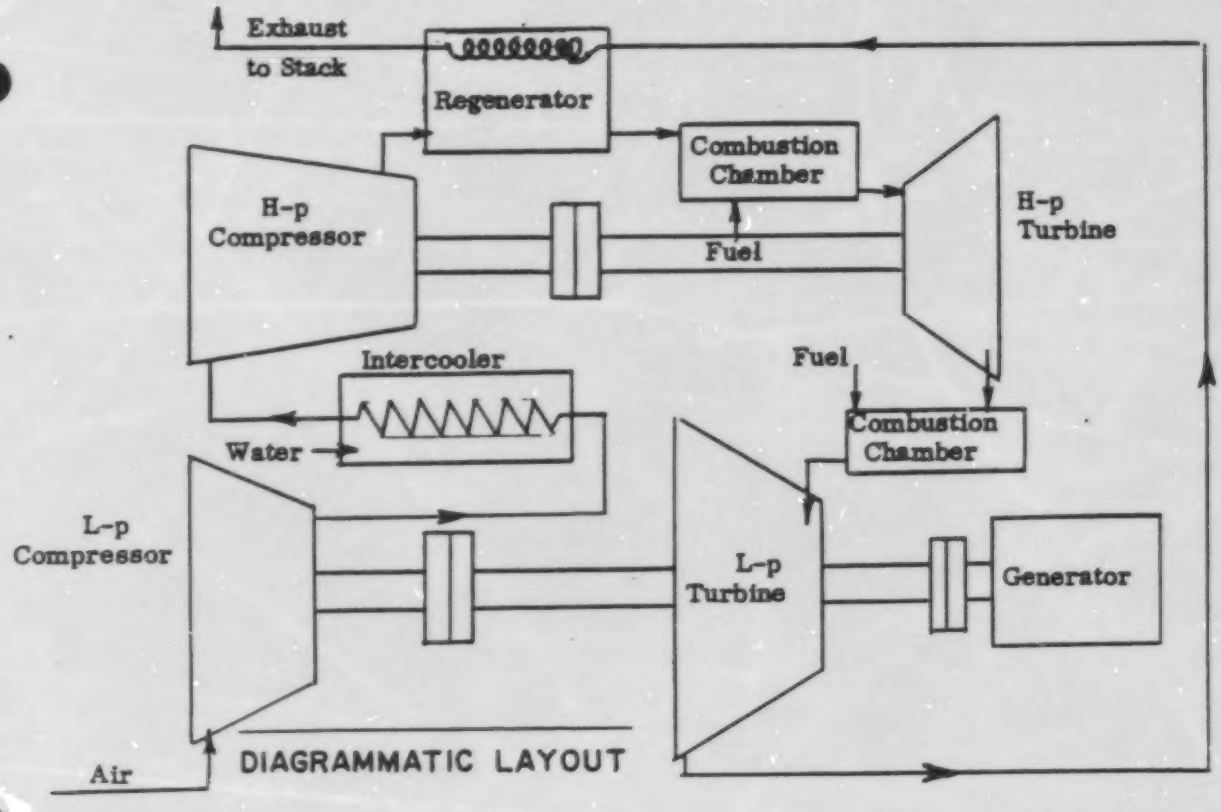
Dual shaft machines are used to improve efficiency in the generation of electricity when operating at part loads. The dual shaft arrangement allows the compressor and turbine speeds to vary while the secondary turbine and generator run at a constant speed. A dual shaft arrangement is shown in the following diagram.



# Information

Regenerators, Intercoolers and Reheaters

The efficiency of a turbine can be increased by the use of a regenerator to capture the thermal valve of the exhaust gas and return it to the inlet air at th combustion chamber. Another efficiency improvement can be made by intercooling the air during compression. This process reduces the work to the compressor. If the air is reheated during its expansion within the turbine, the output of the turbine can be increased. Intercoolers and reheaters are devices used to improve the efficiency of gas turbines. The diagram below shows the arrangement of regenerators, intercoolers and reheaters in a dual shaft turbine.

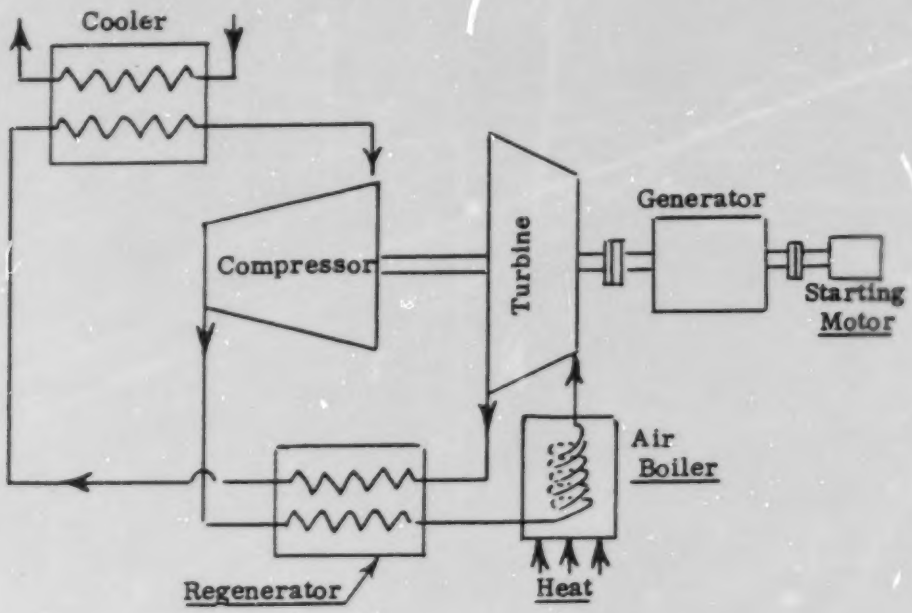



DIAGRAMMATIC LAYOUT

# Information

## Closed Cycle Systems

An open cycle system pulls its air from the atmosphere and dumps it back as exhaust. A closed cycle system circulates the air through its system continuously. Air heaters are required for heating the air that enters the turbine. Coolers are used to cool the air before it enters the compressor. The air heater is the major drawback to the use of closed cycle plants. A diagram of a closed cycle system is shown below.




GAS TURBINE COMPONENTS

## Compressors

Positive displacement compressors pull in air and compress it before releasing it to the turbine. Some type of reciprocating piston or rotary device is used to compress the air. Several types of positive displacement compressors are used in gas turbine operations.

# Information

1. Reciprocating type
2. Lobe type
3. Vane type

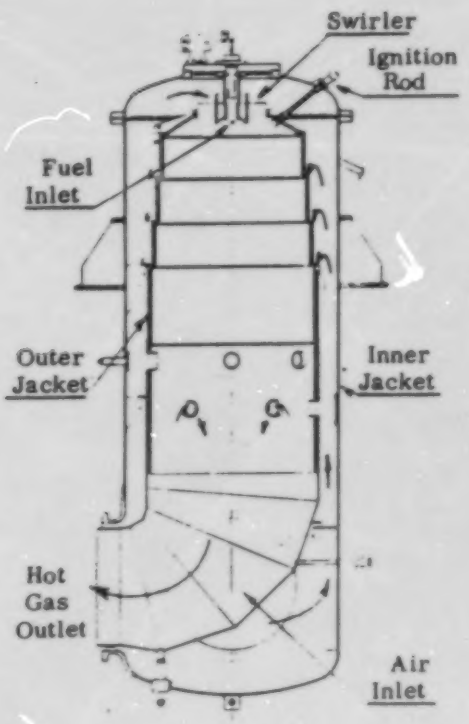
A dynamic type compressor uses rotating blades to squeeze the air into a compressed state. Two types of dynamic compressors are used in gas turbine operation.

1. Centrifugal type
2. Axial flow type

The lobe type compressor is common to gas turbine applications because it can be built in large sizes and with maximum efficiencies.

Combustion Chambers

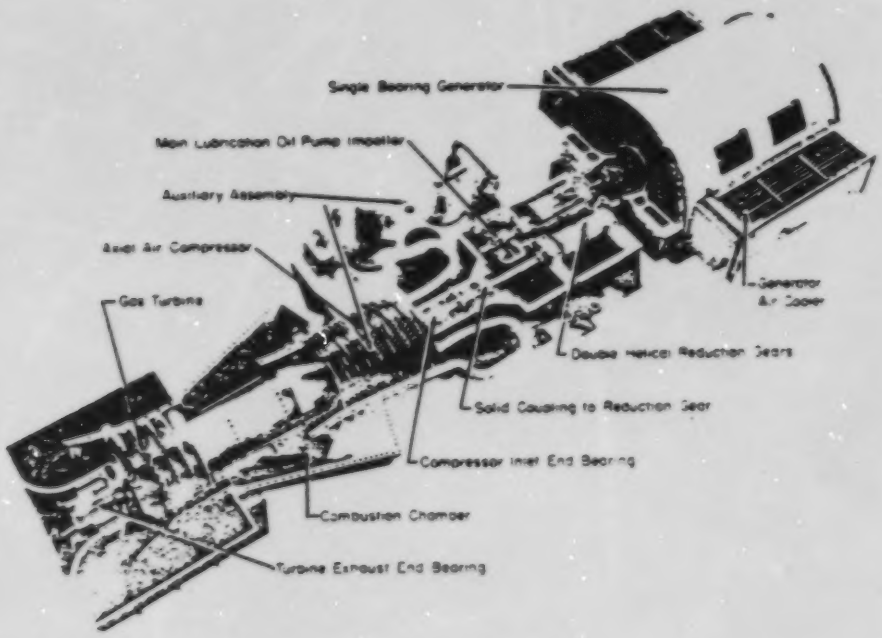
The combustion chamber heats the air between the compressor and turbine. Remember that the air is cooled before entering the compressor and heated before entering the turbine. At the top of the combustion chamber is a swirler and ignition rod. As the air enters near the bottom of the chamber, it moves upward between an inner jacket and an outer jacket. As it moves upward, the air mixes with combustion gases. At the top, part of the air is mixed with fuel by the swirler and is used in the combustion process. The hot gas leaves the combustion chamber and moves into the turbine. Ignition rods are used to light the combustion chamber. The following diagram shows parts of a combustion chamber.



# Information

Turbines

The major difference between gas and steam turbines are in the blading. Blade spacing is greater in the gas turbine because air flow requires fewer stages than steam. Gas turbine blades are subjected to higher temperatures and must be constructed of heat resistance metals. A typical gas turbine and its components are shown below.



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# Assignment

* Complete job sheet.
* Read pages 22 - 34 and 41 - 42 in supplementary reference.
* Complete self-assessment and check answers.
* Complete post-assessment and ask instructor to check answers.

# Job Sheet

INSPECT A GAS TURBINE PLANT

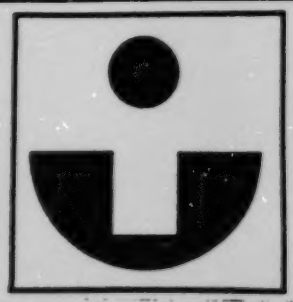
* Locate a gas turbine plant in the community.
* Ask permission to observe the unit.
* Identify the following:
  - Manufacturer of compressor and turbine
  - Single or dual shaft
  - Does it have a heat exchange (regenerator)?
  - Does it have an intercooler?
  - Does it have a reheater?
  - Location of generator
  - Type of compressor
  - Location of combustion chamber
* Ask questions of operator until you fully understand the parts and function of the gas turbine equipment.
* Observe start-up and stopping procedures if possible.

# Self Assessment

Match the following terms and phrases.

| | | |
|---|---|---|
| _____ 1. | Regeneration | A. Temperature of air entering a gas turbine. |
| _____ 2. | Positive displacement compresser | B. Saving exhaust heat for improved thermal efficiency. |
| _____ 3. | 700 C | C. Intercools air during compression. |
| _____ 4. | Lobe | D. Reheats air during expansion within the turbine. |
| _____ 5. | Swirler | E. Continuous circulation of air through system. |
| _____ 6. | Closed cycle system | F. Common type of compressor to gas turbine applications. |
| _____ 7. | Axial flow | G. Uses rotating blade to compress air. |
| _____ 8. | Intercooler | H. Uses reciprocating piston or rotary device to compress air. |
| _____ 9. | Dynamic type compressor | I. Mixes air and fuel in combustion chamber. |
| _____ 10. | Reheater | J. Type of dynamic compressor. |

# Self Assessment Answers



B 1.

H 2.

A 3.

F 4.

I 5.

E 6.

J 7.

C 8.

G 9.

D 10.

# Post Assessment

1. An _____ cycle turbine system takes inlet air from the atmosphere and dumps exhaust air back into the atmosphere.

2. The process of saving exhaust heat is called _____.

3. Dual shaft machines improve efficiency of turbines when operating at _____ load.

4. Reheating of air during its expansion in the turbine can be accomplished by the use of a _____.

5. Cooling of air during compression is accomplished by the use of an _____.

6. A _____ cycle turbine system continuously circulates air through its system without dumping exhaust into the atmosphere.

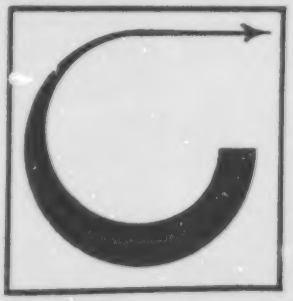
7. List three types of positive displacement compressors.

8. List two types of dynamic compressors.

9. _____ rods are used to light the combustion chambers.

10. The _____ mixes air with fuel at the top of the combustion chamber.

# Instructor Post Assessment Answers



1. Open
2. Regeneration
3. Part
4. Reheater
5. Intercooler
6. Closed
7. Reciprocating, lobe, vane
8. Axial flow, centrifugal
9. Ignition
10. Swirler

# Supplementary References

* Correspondence Course. Lecture 8, Section 3, First Class.
Southern Alberta Institute of Technology. Calgary, Alberta, Canada.

SOUTHERN ALBERTA INSTITUTE OF TECHNOLOGY
CALGARY, ALBERTA

Correspondence Courses
Power Engineering

SECTION 3

PRIME MOVERS

First Class
Lecture 8

## GAS TURBINES

Gas turbines would be chosen as the prime movers for a power plant in cases where the location or purpose of the plant makes some or all of their characteristics particularly advantageous.

These characteristics are:

- Little or no water requirements;
- High power to mass ratio, and consequently only light foundations needed;
- Rapid starting and loading ability with no stand-by losses which makes them ideal for peak load purposes;
- Simple plant layout with very few auxiliaries, hence small location area required;
- Low installation cost, and reduced operating labour requirements;
- Best efficiency and maximum output attained with low ambient air temperatures.

Gas turbines are being used at the present time for electricity generation in sizes up to about 30 megawatts, though generally the heat rate of a gas turbine will be higher than the heat rate of a steam turbo-generator set of equivalent power.

They are, of course, extensively used for aircraft propulsion where they have many advantages over the reciprocating engine such as use of lower grade fuels, less fire hazard, better balance, greater power to mass ratio, simpler cooling problems, etc.

They are also employed in industrial processes, such as iron and steel production where they are used for blast furnace blowers (in this case the gas turbine uses blast furnace gas as fuel); in oil refineries using the Houdry oil cracking process, and in oil pipeline pumping units. They are also used for the propulsion of ships and railway locomotives, and in smaller sizes supplied with exhaust gas and used to drive centrifugal blowers for supercharging internal combustion engines.

Principles of Operation

The principle of the gas turbine was first proposed during the 18th century but its practical application has only been developed in recent years. This has been dependent on the development of efficient compressors and the perfecting of suitable materials to withstand the high temperatures used in the turbine blading.

The gas turbine derives its power from the expansion of a working fluid through the blading of a turbine in a similar manner to the expansion of steam through a steam turbine and uses the same principles of impulse and reaction.

It is interesting to note in this connection that the maximum compression ratio in a gas turbine plant will be about 7 : 1 and this is the pressure ratio which is available to the turbine. A steam turbine plant however may have a pressure of 17500 kPa at the turbine stop valve and 3.5 kPa at the exhaust, which represents a pressure ratio of 5000 : 1.

Gas turbine speeds of rotation vary quite considerably with the size of the machine, for example, a large machine of 30,000 kW may run at 3,600 rev/min, an 6000 kW machine at about 5,000 rev/min, a 4000 kW unit at about 6,000 to 7,000 rev/min, and 750 kW machines from about 10,000 to 20,000 rev/min.

The gross thermal efficiency of simple cycle gas turbines ranges from 17 to 22%.

The working fluid can be any hot gas, but is most often air which has been compressed and then heated in a combustion chamber. The gas turbine drives its own air compressor, the power absorbed by the compressor being about 2/3 of the total turbine power output.

Mention was made in Lecture 1 of the Brayton Cycle, used as the basis of operation of a gas turbine. It was shown that for any given compressor inlet air, and turbine inlet gas temperatures, there is an optimum compressor pressure ratio at which maximum work output would be achieved by the gas turbine, together with the best practical efficiency.

Once the machine has been designed, the compressor pressure ratio is fixed and the variables remaining which affect the machine efficiency and output are the compressor inlet air temperature, the turbine inlet temperature and the mass gas flow.

The compressor inlet air temperature, in the case of open cycle gas turbines, is dependent only on the atmospheric air temperature; the lower this is, the higher will be the gas turbine efficiency and output.

For example, a Brown-Boveri machine rated at 20 MW output and a heat rate of 15,350 kJ/kWh with ambient air temperature of 70°C is quoted as delivering 30 MW at 12,630 kJ/kWh when the air temperature drops to -40°.

The temperature of the gas at the turbine inlet can be controlled by variation of the quantity of fuel burned. Maximum load and maximum efficiency will be obtained with top inlet temperature.

The turbine inlet gas temperature is limited by the material of the turbine blades and is generally not more than 650°C to 700°C for power plant gas turbines. Aircraft engines are designed for a shorter operating life and the inlet gas temperature used in this case is approximately 870°C.

Efforts are being made by designers to enable gas turbine blades to withstand higher gas temperatures through the use of some form of blade cooling. Most of the suggested designs employ the circulation of a coolant through the blades such as water vapourizing to steam, or forced air, or some liquid sealed within the blades rejecting its heat by circulation to another fluid in the turbine wheel.

If the gas inlet temperature could be raised to 950°C, gas turbine thermal efficiencies would advance to the 30% range. The mass gas flow cannot be varied independently in the open cycle gas turbine plant and is a function only of the compressor speed; load variations in this type of plant are carried out by control of the turbine inlet temperature through the quantity of fuel burned. The closed cycle gas turbine design however, does allow control of the mass gas flow through the machine and this is used together with turbine inlet temperature as the method of load control.

The energy available to the turbine for production of work depends upon the heat content of the gases per kg and the mass of gas flowing; the calculation of available power is carried out in a somewhat similar manner to that for the steam turbine.

A typical power calculation will be found at the end of this lecture.

The fuel used in gas turbines up to the present day, has been restricted to oil or gas. Many experiments have been carried out, using coal as fuel, but troubles due to slagging and erosion of the turbine blades have yet to be overcome.

## PLANT ARRANGEMENTS

### Open Cycle Systems

The simple gas turbine plant consists of a compressor, a combustion chamber and a turbine. Fig. 1 shows a diagrammatic layout.

Atmospheric air is compressed and then heated to the maximum cycle temperature by the burning of fuel in the combustion chamber. The resulting products of combustion expand through a turbine and exhaust to atmosphere. The turbine drives the compressor and the balance of power is available to drive the generator. A starting motor is required to get the system into operation. Fig. 2 is a descriptive sketch of the plant.

### Regeneration

The exhaust gas temperature from the simple gas turbine will be very high. The thermal efficiency can be appreciably improved by the addition of a heat exchanger to transfer some of this heat in the exhaust gases to the air prior to its entry to the combustion chamber. This is termed regeneration. Fig. 3 shows this arrangement diagrammatically.

The optimum sized regenerator recovers about 75% of the exhaust heat in the gas which is available above the compressed air temperature. The use of a regenerator will make typical reductions in the exhaust gas temperature from 450 to 260$^{o}$C while increasing the compressed air temperature from 230$^{o}$C to 400$^{o}$C.

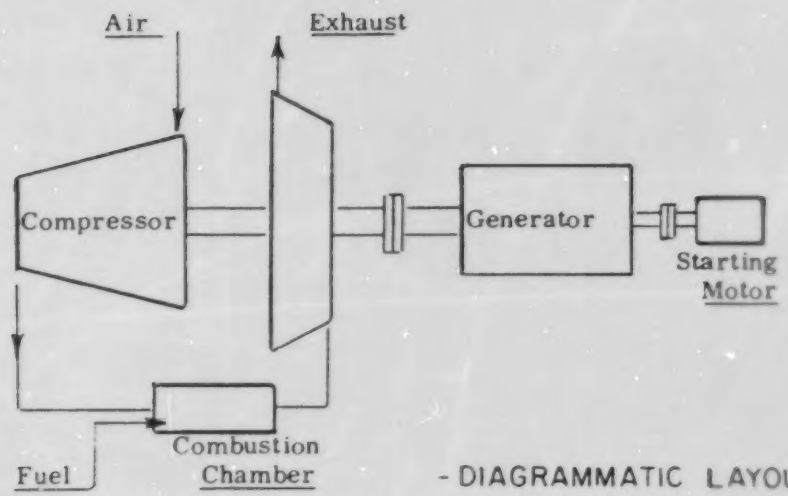
Fig. 4 is a descriptive sketch of the plant showing the addition of the heat exchanger. The use of a heat exchanger or regenerator in a gas turbine means that a large area of heat transfer surface has to be supplied. The physical bulk of a regenerator can be accommodated in a stationary power plant but is impractical for aircraft or locomotive engines.

In Lecture 1 of this Section, the ideal Brayton cycle was described and Fig. 23 was drawn to represent this cycle on a T$\Phi$ diagram. Fig. 5 herein is a similar diagram. If the gases are passed out to waste at point 4 (the turbine exhaust) the rejected heat will be that shown by the shaded area under 4 - 1. If a regenerator is ued to recover some of this waste heat then the air temperature can be raised after leaving the compressor to a figure approaching that of the turbine exhaust gas.

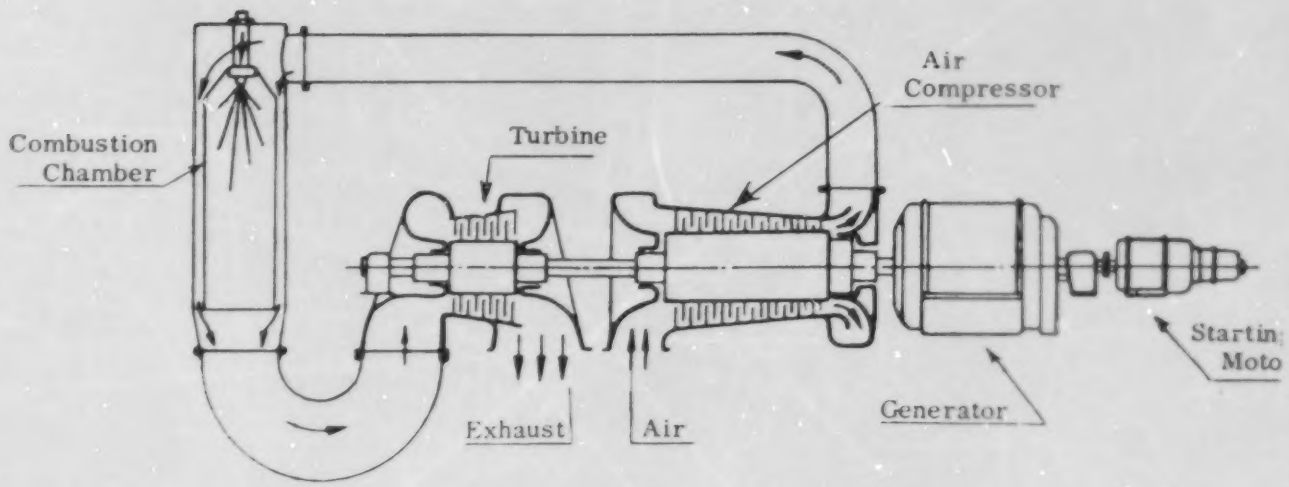
Fig. 6 illustrates the principle. The area under 4 - 4' represents heat given to the regenerator and used to raise the temperature to 2'. The heat rejected to exhaust is now only that represented by the shaded area under 4' - 1.

This makes a considerable improvement in the gas turbine thermal efficiency. For example, a simple gas turbine with reasonable component efficiencies, compressor 84%, turbine 86%, combustion 97% and a gas temperature of 650$^{o}$C, but without a heat exchanger will have a thermal efficiency of the order of 20%. If a heat exchanger of 75% effectiveness is added this will be raised to about 25% (75% effectiveness meaning, able to transfer to the air 75% of the heat available in the exhaust gas).

PE1-3-8-4



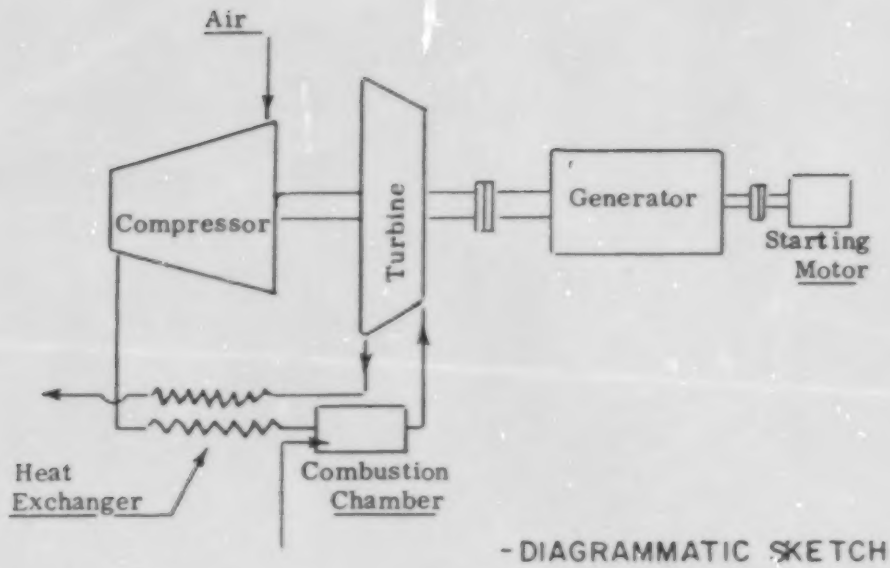

- DIAGRAMMATIC LAYOUT -
Fig.1




- DESCRIPTIVE SKETCH -
Fig.2

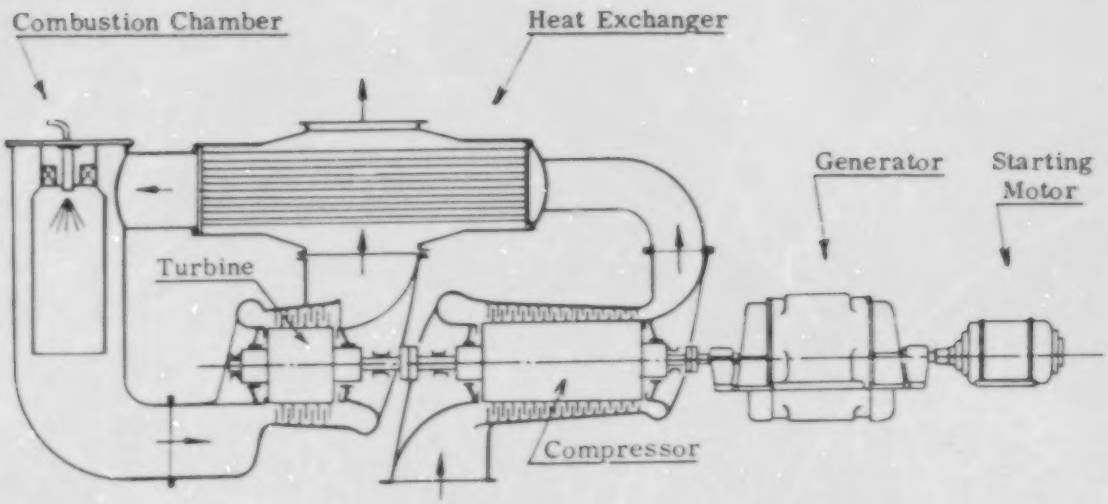
## SIMPLE OPEN CYCLE GAS TURBINE PLANT

PE1-3-8-5




- DIAGRAMMATIC SKETCH -

Fig 3




- DESCRIPTIVE SKETCH -

Fig 4

OPEN CYCLE

GAS TURBINE with REGENERATOR

PE1-3-8-6

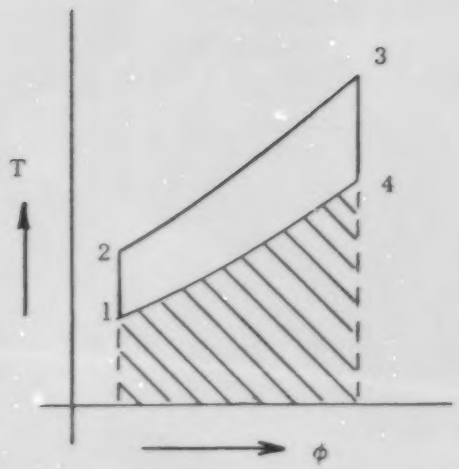



Fig. 5

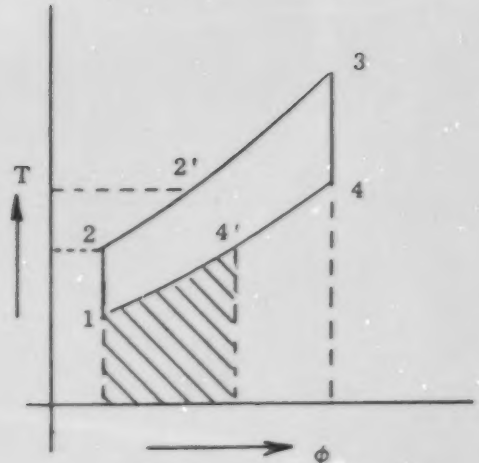



Fig. 6

Tφ Diagram for Brayton Cycle

Attractive as these gains appear, many industrial gas turbine plants in operation today do not include regenerators, for the reason that the plants are designed essentially to supply peak load demands. They are intended to be run for short periods only and are most attractive in their simplest form and with the minimum cost of construction and installation. The heat exchanger or regenerator is necessarily a large and expensive item of plant equipment since the pressure losses in both air and gas flows must be kept to a minimum.

Dual Shaft Machines

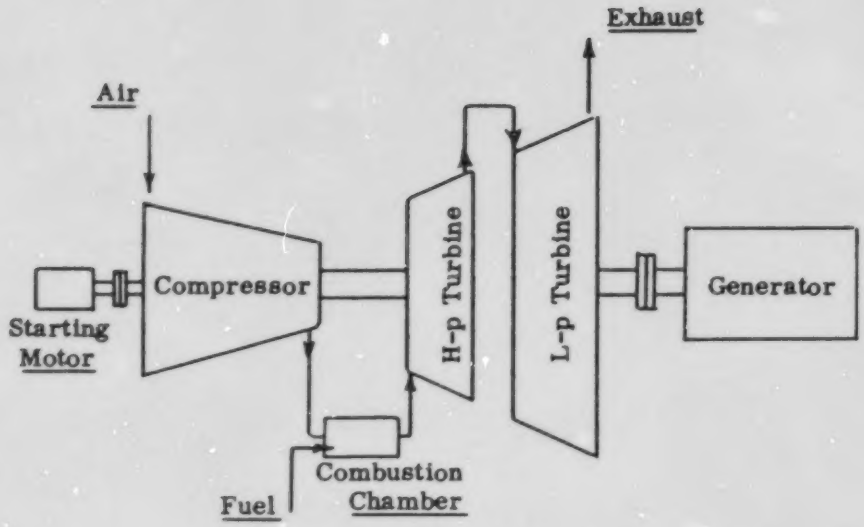
The single shaft gas turbine is essentially a constant speed machine, variations in speed causing considerable changes in the power output. For example, with the turbine inlet temperature kept constant at its top figure a 10% reduction in turbine speed will reduce the power output by about 25%, and a 25% reduction in speed makes about 60% reduction in the power output.

When the shaft is separated as shown in Fig. 7, the unit becomes a dual shaft machine.

When used with a gas turbine plant for electricity generation this arrangement gives better efficiency at part loads by allowing the speed of the compressor and primary turbine to vary, while the secondary turbine and generator remain running at synchronous speed. It also reduces the power required for the starting motor as this does not now drive the secondary turbine and generator.

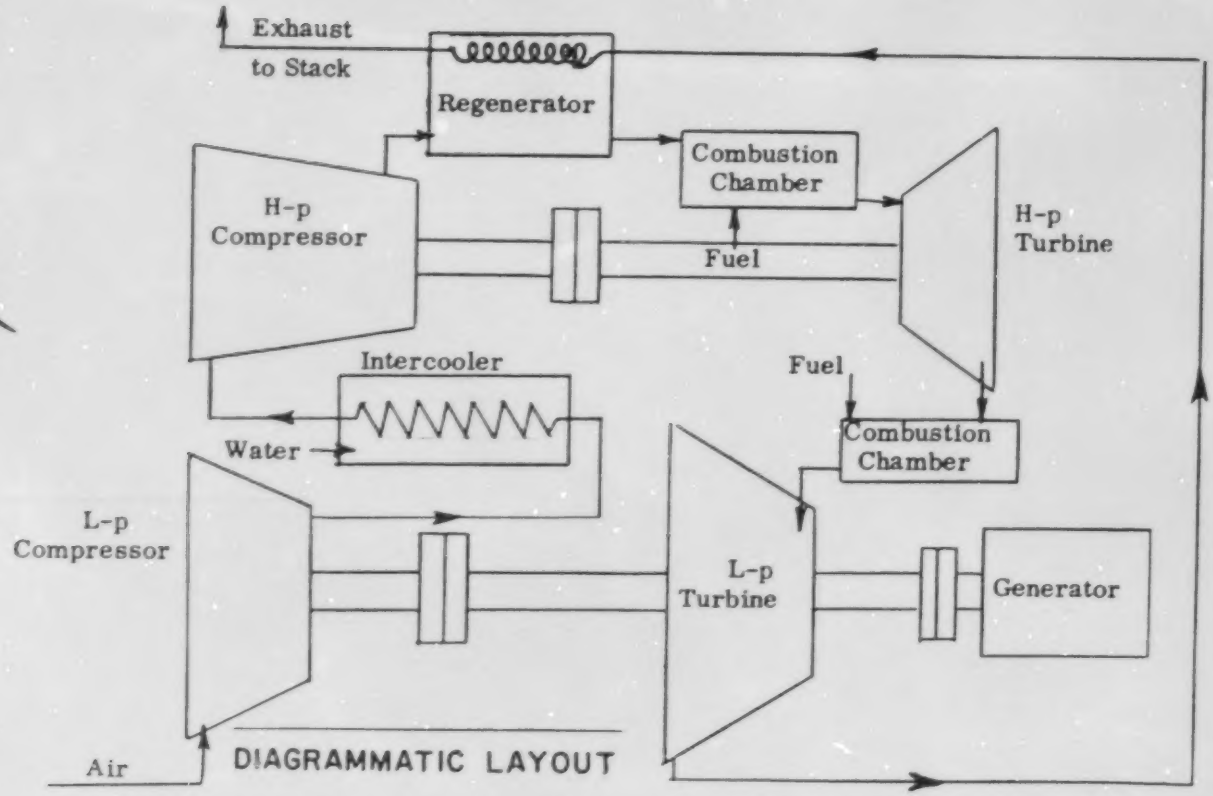
It can be adopted for gas turbines used for mechanical drive in which case it can give variable speed operation of the power turbine take-off and its driven machinery.

PE1-3-8-7




OPEN CYCLE GAS TURBINE with DUAL SHAFT

Fig. 7




OPEN CYCLE GAS TURBINE with REGENERATOR, INTERCOOLER and REHEATER

PEI-3-8-8

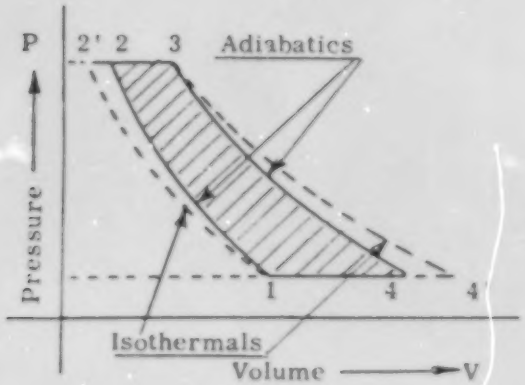
Fig. 8

Intercooling and Reheating

Further improvements in the gas turbine efficiency and output can be obtained if the air can be intercooled during compression and the gas can be reheated at some stage in its expansion through the turbine.

Fig. 8 shows the arrangement with Regenerator (or heat exchanger), Intercooler and Reheater. Now the machine is in two-shaft form with two separate compressors and turbines and a second combustion chamber for reheating the gas between the two turbines.

Referring to the original PV diagram for the Brayton Cycle (Fig. 22, this Sect. Lecture 1 - repeated as Fig. 9 below): the cross-hatched area should be the amount of work available for the cycle. Compression 1 - 2 and expansion 3 - 4 were each adiabatic operations. If these could have been carried out as isothermal (constant temperature) operations, the compression would be 1 - 2' and the expansion 3 - 4'. This would have increased the available work considerably.




P-V Diagram for Brayton Cycle
Fig. 9

An approximation can be made to this condition by intercooling the air at one or more stages in the compression and by reheating the gas at one or more stages in the expansion. This is indicated on Fig. 10. The intercooling reduces the work input to the compressor and the reheating increases the work output from the turbine. The net effect of the whole is to increase the output of the gas turbine and also to increase its thermal efficiency.

For example, General Electric Co. quote a thermal efficiency of 29.1% for a gas turbine of their design driving a 26 MW generator and operating on a regenerative, intercooled, reheat cycle.

PE1-3-8-9

## Uses of Exhaust Heat

One of the largest sources of loss in the gas turbine plant is in the high temperature exhaust gas. The temperature at the turbine exhaust may be 425°C. The use of a regenerator has been mentioned as one method of recovering some of this heat. A waste heat boiler producing steam for space heating or process work is sometimes used, either separately or in conjunction with a regenerator.

The fuel burned in the combustion chamber of the gas turbine uses only a small proportion of the total air flowing in the system for combustion (about 20%) so that the oxygen content of the final turbine exhaust gas may be about 17% as against 21% contained in atmospheric air by volume.

Further, the air quantities used by a gas turbine are considerable (about 0.4 $m^3$/min for each kW produced), so this turbine exhaust gas can be used as heated air for combustion in a steam boiler furnace. This combination of plants gives an improved overall thermal efficiency.

A combined steam and gas turbine plant may also be achieved by discharging the compressed air from the gas turbine air compressor into the steam boiler furnace and then expanding the hot exhaust gases from the boiler through the gas turbine. This method results in a supercharged boiler and if some of the gas turbine exhaust heat can be recovered in feed heaters, it will give a greater increase in overall thermal efficiency than the first mentioned combination steam-gas turbine plant. A gross thermal efficiency of 42% has been obtained from this type of combined steam-gas turbine plant.

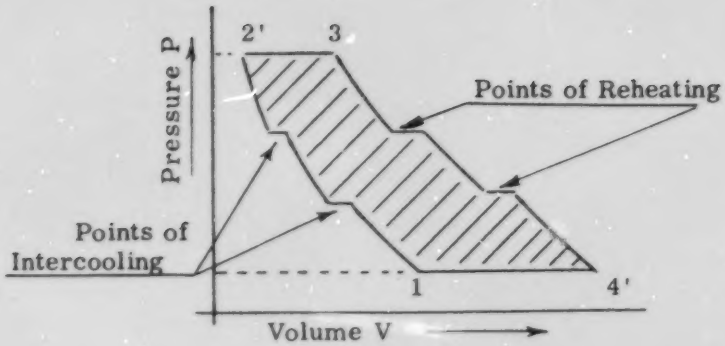
Further mention of combined steam-gas turbine plants will be found in the next lecture.

## Closed Cycle Systems

Note that all of the systems so far mentioned operate on an Open Cycle. That is, the air used is drawn from atmosphere and the exhaust from the turbine is rejected to atmosphere. Almost all of the gas turbine plants in use today operate on the Open Cycle system.

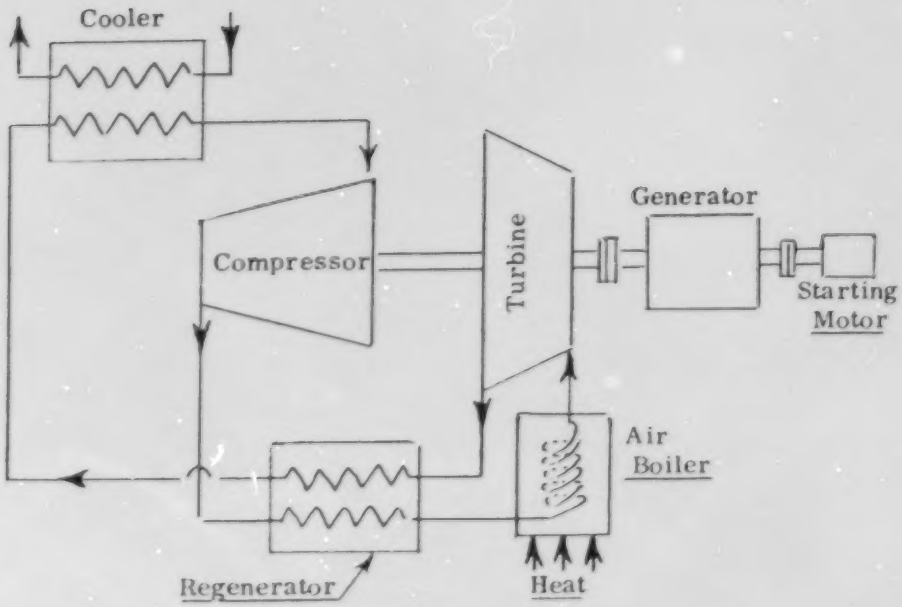
Gas turbines can also be constructed using a Closed Cycle, in which the working medium (air) is circulated continuously through the system.

Fig. 11 shows a layout of the Closed Cycle.




P-V DIAGRAM showing EFFECT OF INTERCOOLING and REHEATING

Fig. 10




CLOSED CYCLE GAS TURBINE PLANT

- DIAGRAMMATIC SKETCH -

Fig. 11

PE1-3-8-11

In this case the working air has to be heated in an Air Heater before entry to the turbine and cooled before entry to the Compressor.

The closed cycle system has many advantages and is capable of full load thermal efficiencies approaching 35%; part load efficiencies remain high also, being up to 30% at 25% load. The added complications however tend to offset the advantages by increasing the cost of installation and maintenance and there are very few plants of this type in commercial operation today.

Since the working air is not drawn into the compressor from the atmosphere, the pressures throughout the system can be made much higher than in the open cycle plant. The turbine output depends upon the mass of gas flowing through it so that the increased air density results in a smaller machine for a given output rating.

Continuous recirculation of the working air with no addition of combustion products avoids fouling of turbine blades and heat transfer surfaces.

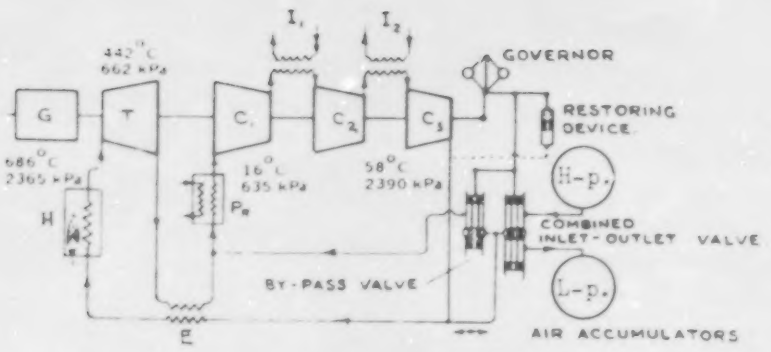
Use of an air heater instead of a direct combustion chamber allows the burning of fuels which would be unsuitable for the gas turbine blading.

Control of the plant output can be effected by variation of the density of the working air; this is done by increasing or decreasing the quantity of air in the system. By this means the speed and working temperatures can be kept constant during load variations and high efficiencies can be maintained at part loads.

The open cycle plant must use air if fuel is to be burned in the combustion chamber; the closed cycle plant however, could be charged with any gas which possesses satisfactory operating characteristics. Hydrogen, Helium, or a mixture of Helium and $CO_2$ have been suggested, the main advantage to be gained over air being the higher heat conductivity. This in turn reduces the necessary heat transfer surfaces required in the air heater, the regenerator and the cooler.

The chief disadvantage of the closed cycle system is the use of the air heater with its large bulk, inefficient heat transfer, cost and design difficulties, in comparison with the direct combustion chamber of the open cycle plant.

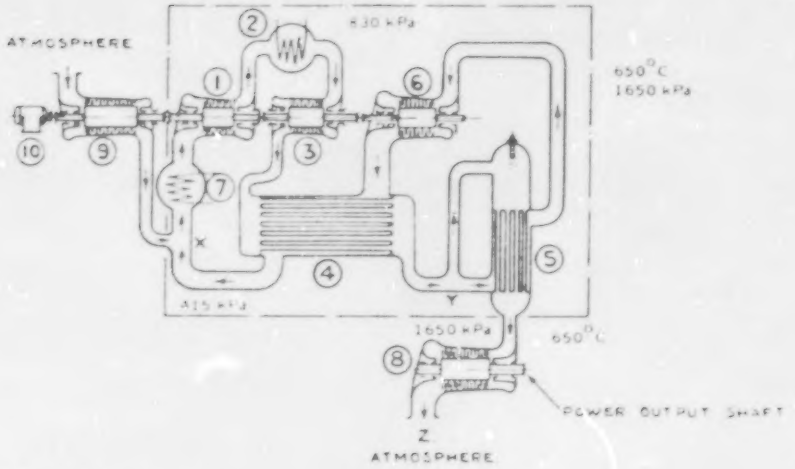
Fig. 12 shows a schematic diagram of a closed cycle gas turbine plant by Messrs. Escher Wyss, and Fig. 13 shows a semi-closed cycle plant developed by Messrs. Sulzer Bros. The semi-closed cycle uses direct combustion gases for the work turbine and thus reduces the size of air heater required.




| | | | |
|---|---|---|---|
| G | GENERATOR | $I_1, I_2$ | INTERCOOLERS |
| T | TURBINE | H | AIR HEATER |
| $C_1$ | L-p. COMPRESSOR | $P_c$ | PRECOOLER |
| $C_2$ | I-p. COMPRESSOR | H-p. | HIGH PRESSURE RECEIVER |
| $C_3$ | -p. COMPRESSOR | L-p. | LOW PRESSURE RECEIVER |
| | | E | HEAT EXCHANGER |

CLOSED CYCLE TURBINE

Fig. 12




| | |
|---|---|
| (1) L-p COMPRESSOR | (6) COMPRESSOR TURBINE |
| (2) INTERCOOLER | (7) PRE COOLER |
| (3) H-p COMPRESSOR | (8) WORK TURBINE |
| (4) HEAT EXCHANGER | (9) COMPRESSOR SUPPLYING COMBUSTION AIR |
| (5) COMBUSTION CHAMBER | |
| (10) STARTING MOTOR | |

SEMI-CLOSED CYCLE TURBINE

Fig. 13

PE1-3-8-13

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Water Injection to Gas Turbines

It was mentioned at the outset that in choosing a type of plant for the production of power, the engineer would be influenced by the reduced first cost, simple layout, suitability for load peak purposes, etc., of the gas turbine plant, despite its lower efficiency and higher fuel costs. It follows that he would be interested in any device which would increase the power output of the gas turbine even at the expense of a reduced efficiency. Especially if this need be used only at peak load periods.

By injecting water into the hot gases before they enter the turbine, a gas turbine output can be increased by about 40% at the expense of about 25% increase in the heat rate. When this is done the water must not contain more than 1ppm of impurities if turbine blade deposits are to be avoided.

The effect of this water injection is that the water is evaporated into steam which has the effect of increasing the density of the working gas in the turbine and hence increasing the turbine output. This is done without any increase required in the work input to the compressor.

The steam leaves the turbine exhaust at high enthalpy however, and this causes the reduction in plant efficiency.

Similar results can be achieved by injecting steam into the working gas or by allowing the hot air leaving the compressor to evaporate water and carry the water vapour with it into the regenerator, combustion chamber, and turbine. Alternatively water can be injected at an intermediate stage of the air compressor where it serves also for intercooling. This is the method generally used in aircraft engines during take-off.

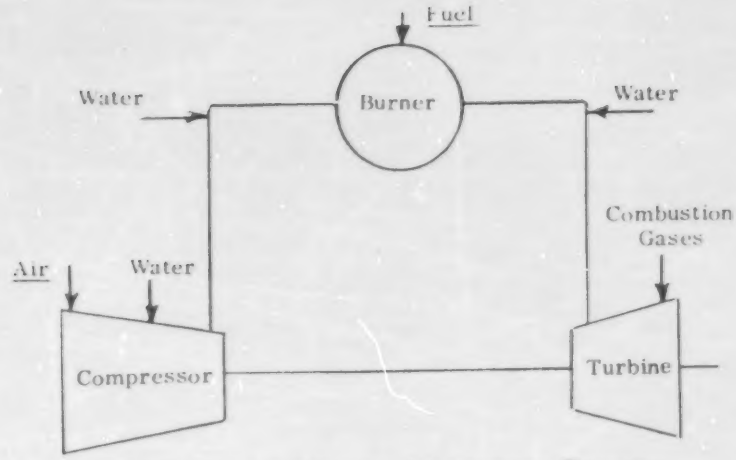
Fig. 14 shows these alternative water injection points diagrammatically.

Free Piston Gas Generators

Reference was made earlier to the fact that a gas turbine can use any hot gas as its working medium and this principle is made use of by combining a gas turbine with a Free Piston Gas Generator. The generator takes the place of the air compressor and combustion chamber in the open cycle plant and supplies hot compressed gas to the turbine. It operates in a similar manner to a diesel engine and burns similar fuels; the pistons are 'free' in that they are not connected to a crank shaft but free to vary the stroke to match the load variations. They are, however, connected to each other by a linkage so as to ensure that they remain in synchronism.

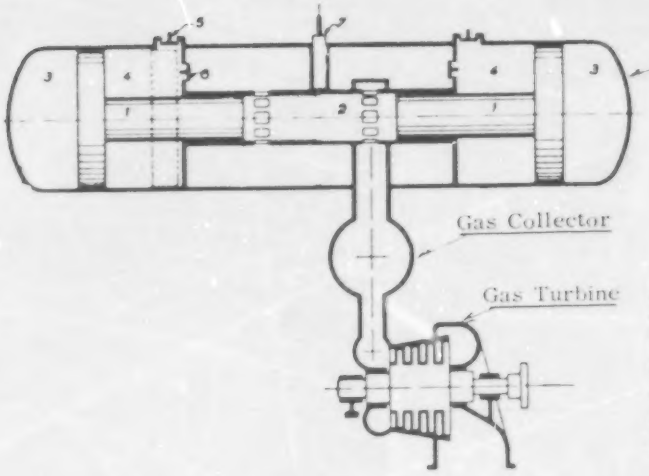
Fig. 15 gives a section view of the plant and Fig. 16 (a), (b), and (c) shows the operation.

The gas generator uses liquid fuel in much the same manner as a diesel engine, the synchronizing linkage mentioned above often being used as the fuel pump driving medium. The generator is started with compressed air, the pistons are set at their outer limits, and a shot of starting air injected into both cushion cylinders. This drives the pistons to the centre and compresses an air charge in D, fuel is injected and the generator is immediately running at normal speed.




GAS TURBINE WATER INJECTION POINTS

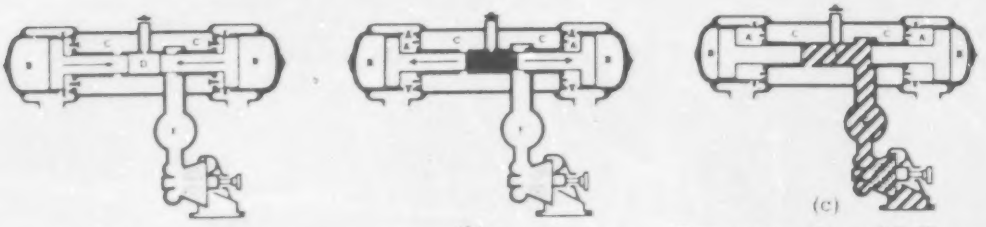
Fig.14




(1) Piston
(2) Engine cylinder
(3) Cushion cylinder
(4) Compressor cylinder
(5) Suction valves
(6) Delivery valves
(7) Fuel injector.

FREE PISTON GAS GENERATOR and GAS TURBINE

Fig.15



(a)

Air in the cushion cylinders (B) forces the pistons together. The compressor pistons force air into the engine case (C), a fresh charge of air is being compressed in the combustion cylinder (D).

(b)

Combustion forces the pistons apart. Air is being compressed (B) and the air pumps (A) are taking in a fresh charge of air.

(c)

The air pumps (A) are full. The ports in the combustion cylinder are uncovered and scavenge air forces the exhaust gases into the gas collector (E) and through the turbine. Air in the cushion cylinders is fully compressed to bounce the pistons back for a fresh cycle.

DIAGRAMS of GAS GENERATOR OPERATION

PE1-3-8-15

Fig.16



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The pistons in a normal diesel engine are connected mechanically to the driving mechanism and hence their speed of movement will depend upon the amount of fuel burned and the load upon the engine.

In the gas generator the pistons are free from mechanical connections and their speed of oscillation depends only upon the pressures in the combustion and cushion cylinders. This can be thought of as similar to a weight suspended from a spring where the frequency of "bounce" will be constant and only dependent on the spring strength.

The load output from the turbine is varied by controlling the fuel quantity injected into the gas generator.

This combination machine has several outstanding advantages: it dispenses with the compressor and combustion chamber of the conventional gas turbine and operates at a higher compression ratio (about 8 or 9 : 1). The overall efficiency obtainable is approximately 40%. It has a minimum of moving parts and therefore low maintenance costs. It is well balanced and requires only light foundations.

The disadvantage is that it is limited in output by the gas volume which can be handled by the "non-flow" part of the system constituting the reciprocating gas generator, the maximum of those built to date being about 1000 kW. This can be overcome by applying a number of gas generators to one gas turbine since the "steady flow" type of system of the gas turbine has no gas volume limitation. For example, six gas generators supplying one turbine have been constructed and used for ship propulsion.

## GAS TURBINE CONSTRUCTION

### Compressors

It was stated in the Principals of Operation, Page 2 of this lecture, that the use of the gas turbine as a practical prime mover was dependent upon the development of high efficiency compressors. The reason for this is that the proportion of the gas turbine output used up in driving the compressor is so large that the compressor must operate efficiently in order to achieve a useful net output of power from the plant. In present day plants, about two-thirds of the total turbine output is absorbed in driving the compressor.

Compressors can be divided into two general types, namely the Positive Displacement type and the Dynamic type.

The Positive Displacement type draws in a quantity of air, traps it, and then compresses it by means of a reciprocating piston or some rotary element before discharging against the outlet pressure. This type includes reciprocating compressors, lobe type rotary compressors, and vane type compressors.

The Dynamic type imparts a high velocity head to the air by means of rotating blades and then converts this to a pressure head before discharging. This type includes centrifugal compressors and axial-flow compressors.

Of these compressors the centrifugal, the axial-flow and the lobe type rotary have been used with gas turbines. The multi-stage axial flow compressor is the type most generally used in plants designed for power production and will be described first.

Axial-flow Compressors

This type of compressor operates on a principle similar to a turbine, but acting in reverse. The moving blades act upon the air so as to increase its velocity and discharge it axially into the next row of fixed blades, rather as though each moving blade was a small section of a propeller. The fixed blades tend to slow the air down in its passage through them and so raise its pressure.

If the moving blades are properly shaped they will cause the air to be compressed in its passage through them so that compression takes place in both fixed and moving blading. If the pressure rise in each is equal, the compressor is said to be symmetrically staged and is similar to a reaction turbine (in reverse).

Fig. 17 is a plan view of a 6700 kW single shaft gas turbine built by Clark Brothers Co. of N.Y. and shows a typical multi-stage axial flow compressor.

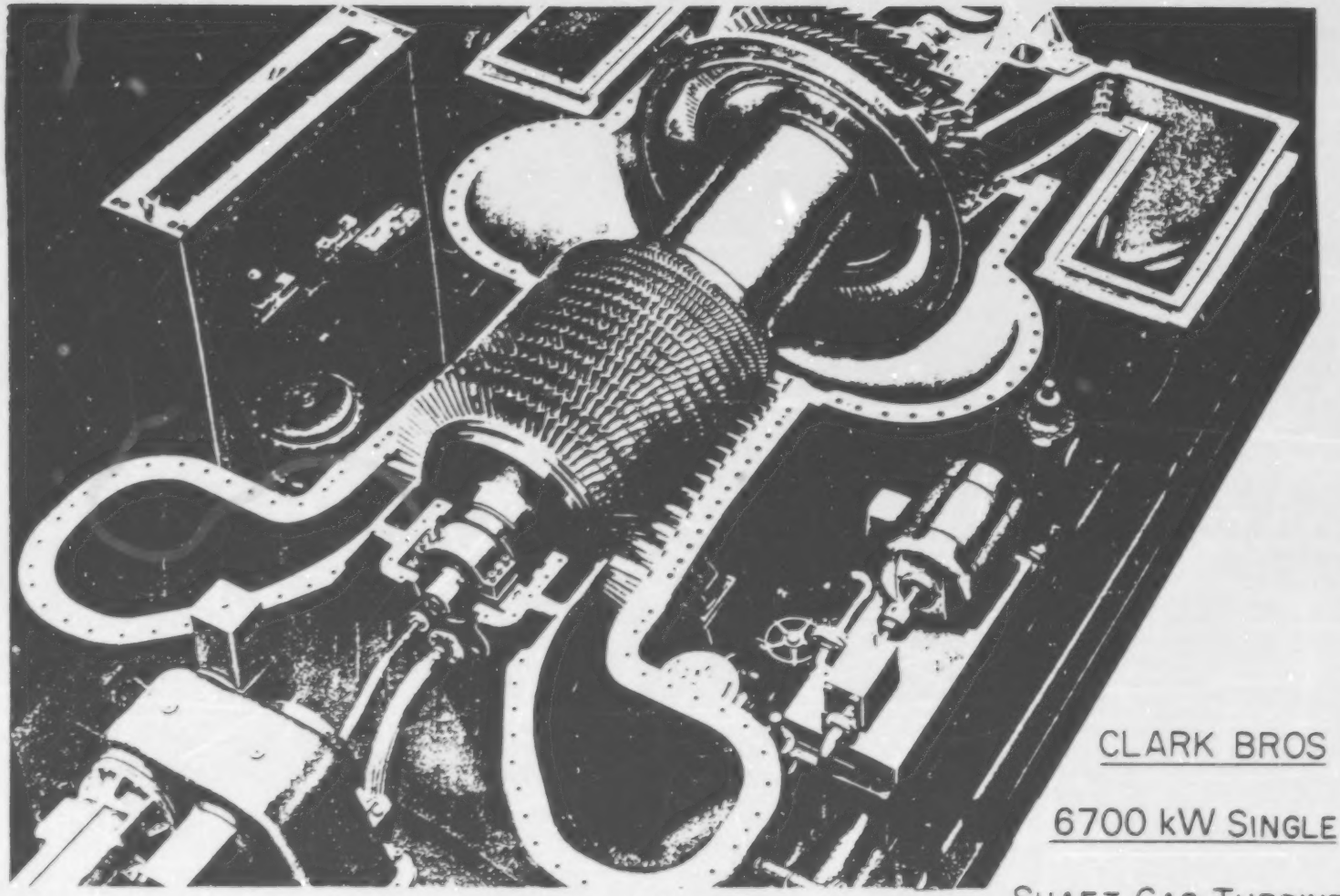
Unlike the centrifugal compressor, the pressure increase in each stage of the axial compressor is quite small. The pressure ratio per stage is about 1.1 or 1.2 : 1 but the efficiency of the axial compressor is about 80% to 85% which is higher than the centrifugal.

The axial compressor has the disadvantage that its discharge pressure becomes unstable at low output and this can cause severe surging with possible damage to compressor and turbine.

Centrifugal Compressors

This type (sometimes called radial-flow compressors) operates on a principle similar to that used by the centrifugal pump. The suction is taken into the centre of an impeller and discharges from the periphery.

Fig. 18 shows a two-stage compressor manufactured by Elliott Company.



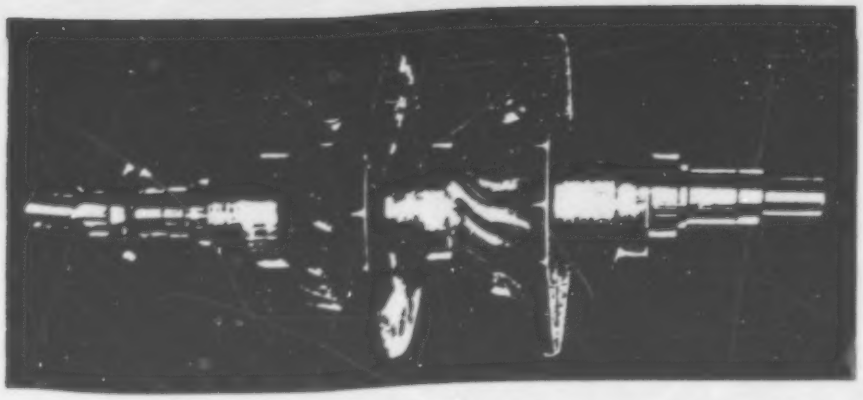
CLARK BROS

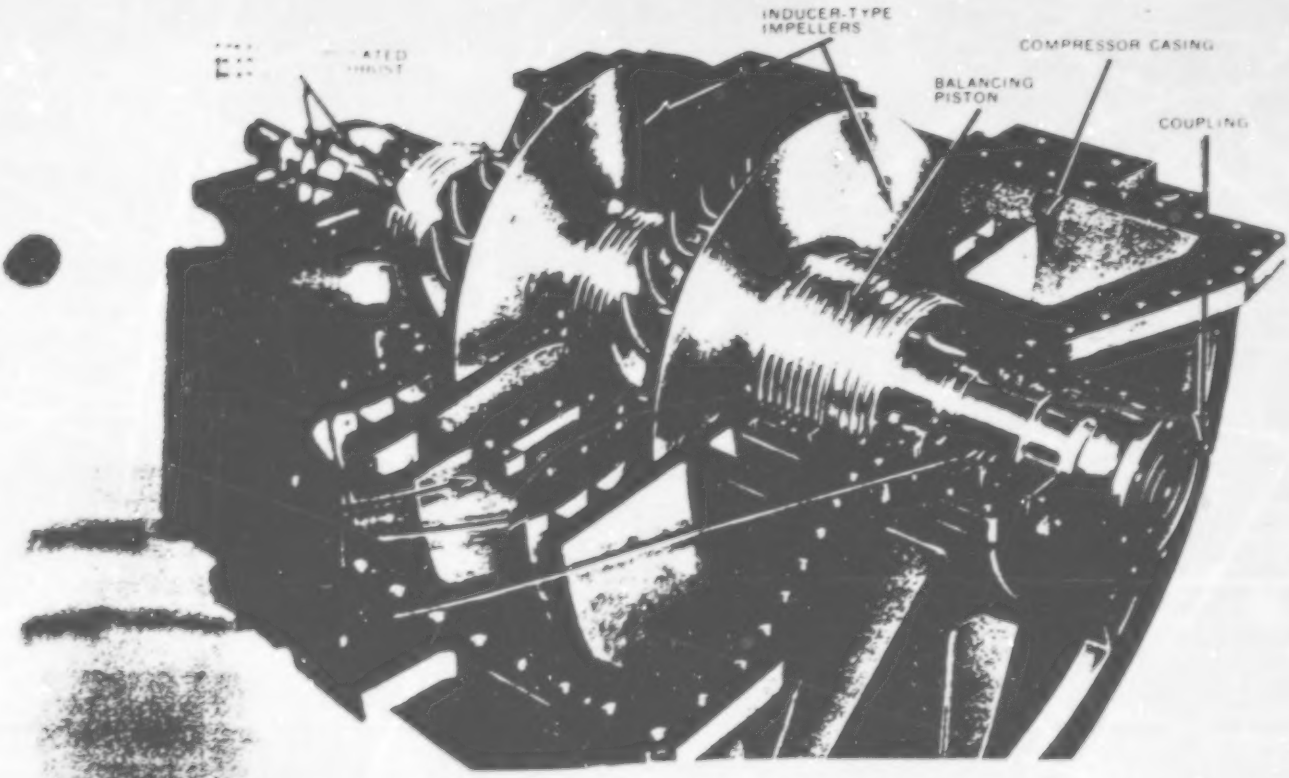
6700 kW SINGLE

SHAFT GAS TURBINE



Fig 17






ELLIOTT CO TWO-STAGE COMPRESSOR

Fig 18

PKT-3-8-19

Their principal advantages are simple and rugged construction, short length, high pressure ratio per stage (can be up to 6 : 1) and the fact that their discharge pressure remains stable over a wider range of loading than the axial-flow compressor.

Their disadvantages are that when built in multi-stage form they are very bulky, and suffer pressure losses in the inter-stage passages. Their efficiency is less than the axial type, being only about 70 to 75% and these disadvantages are sufficient to make the axial type preferable for gas turbine power plant work.

Positive Displacement Compressors

The reciprocating compressor does not find application with the gas turbine because it is limited in the volume which it can pass in any given time, but it has a very desirable characteristic in that although its discharge pressure may vary with the quantity of air drawn in and with the compressor speed, it will always remain stable. It will not surge under any condition of loading.

This characteristic is also found in the lobe type of rotary compressor and is the principal reason for its use with gas turbines when large variations of load or speed have to be provided.

Fig. 19 shows the principle of the Roots and Lysholm compressors. The Roots compressor, Fig. 19(a), has rotors rotating in opposite directions within a casing, air is carried round from inlet to discharge in the spaces between rotors and casing in the manner of a gear pump. The air flow is transverse, across the axis of the rotors.

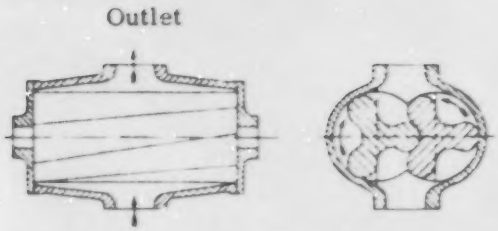
The Lysholm compressor, Fig. 19(b), has two rotors with lobes arranged helically on each. The rotors mesh very closely with each other without actual contact and the clearance between rotors and casing is kept very small.

In rotating, the rotors draw air from the inlet end of the casing and compress it in the decreasing spaces between the lobes as it passes axially towards the discharge end.

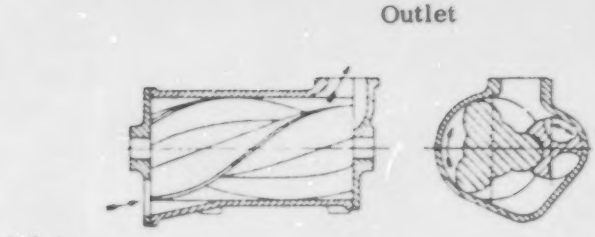
These compressors can be built in large sizes and have high efficiencies (80 - 85%) together with stable operation over a wide range of loading.

Combustion Chambers

The combustion chamber, or combustor, in the open cycle gas turbine is used to heat the working air after its discharge from the compressor and before entry to the gas turbine. It must do this with a minimum loss of pressure and with the minimum of combustion impurities since these will be carried with the air into the turbine blading.




ROOTS COMPRESSOR
(a)




LYSHOLM COMPRESSOR
(b)

Fig. 19

About 20 per cent of the air entering the combustor is mixed with the fuel in the flame tube as combustion air; the remainder - 80% - flows on the outside of the tube and services as cooling air.

The temperature of the burning gases in the tube will be 1370°C to 1650°C but the final mixture of the air and hot gas leaving the combustor is limited to the temperature that the turbine blading can withstand over its working life. This is about 650 to 700°C in present day practice so that the cooling air and hot gas must be thoroughly mixed before leaving the combustor.

Some of the gas turbine designs use a single, large volume combustor and others a series of smaller combustors disposed radially around the engine between the compressor and the turbine.

Generally the large combustion chamber will be used when a regenerator is included in the plant or where heavy oil is to be the fuel used.

Fig. 20 shows a section of a combustion chamber of the single type as used by Brown Boveri. The air inlet is located low down and the air flows upwards between the inner and outer jackets. Approximately halfway up some of the air is mixed with the combustion gases through adjustable mixing nozzles; the remaining air serves to cool the telescopically arranged cylindrical sections forming the inner tube, finally flowing through the annular spaces between sections. Almost 20% of the total inlet air reaches the top of the combustor and enters the swirler to act as combustion air for the fuel. An electrically heated ignition rod is positioned close to the swirler.

Fig. 21 illustrates a section through a combustor used by Associated Electrical Industries (Canada) Ltd. Six of these combustors are used on a machine of 6.5 MW output burning natural gas or distillate oil.

Each combustor is made up of an inner chamber, which is carried on radial pins to allow relative expansion, and an outer casing. Interconnecting pipes are provided between the six combustors to give uniform combustion conditions and to carry the flame from one to the other during the starting sequence, only two of the combustors carry igniter elements.

Turbines

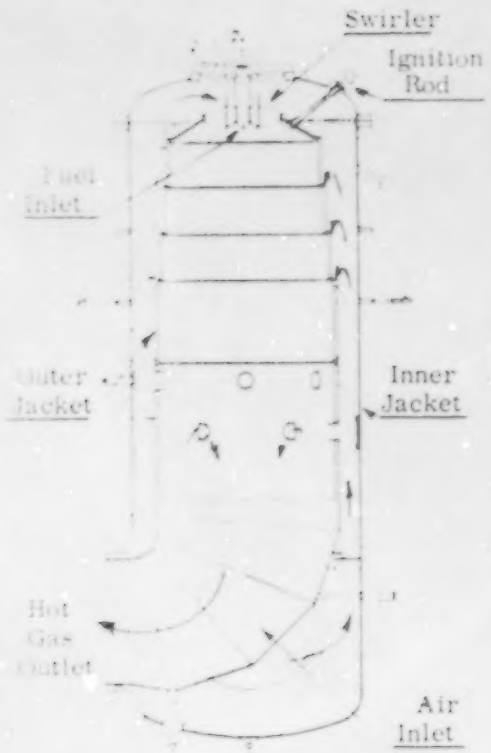
Gas turbines use impulse and reaction blading, working on the same principles as the blading in steam turbines. The main difference between gas and steam turbine blading arises from the difference between the working medium. Gas turbines have a much less number of stages because the total pressure drop available is relatively small. The spacing of the blades is much greater because of the high volume flow.

The stresses in the turbine rotors and blading are high because of high gas temperatures. In order to withstand this, the rotors are made from heat resisting steel, and owing to the difficulty of making large forgings of this material the rotors are generally made up of discs bolted or welded together.

Fig. 22 shows a section through a Brown Boveri gas turbine set in which both the gas turbine and the axial flow compressor rotors are made up of welded discs.

Fig. 23, (a) and (b), show a Westinghouse Gas turbine rotor of bolted construction.

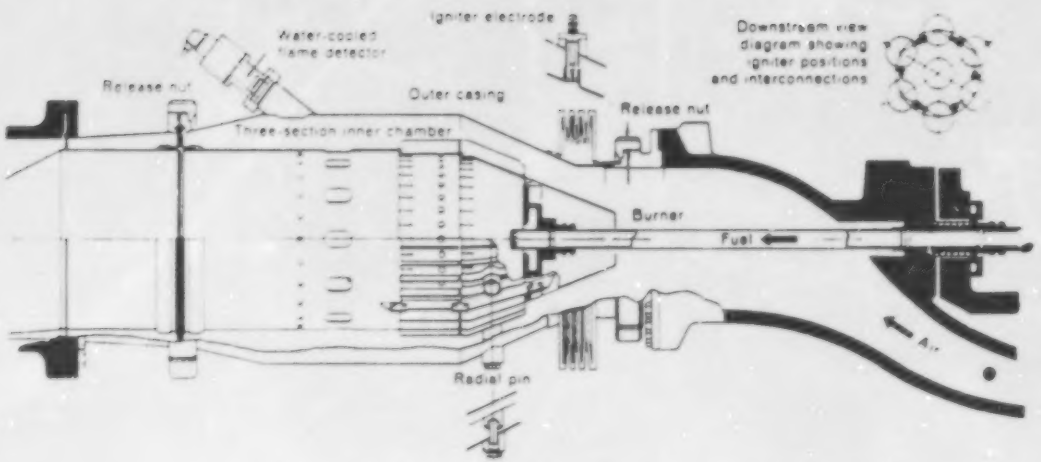
Gas turbine blading is made of heat resisting steel, forged and machined to shape. Steps are taken in some designs to cool the blading, using hollow blading with some coolant such as compressed air flowing through. Figs. 24 (a) and (b) show a turbine moving blade and a turbine fixed blading half diaphragm respectively, manufactured by A.E.I. Co.




Air inlet and gas outlet are located at low level; the adjustable openings for the mixing air are in the centre; the fuel nozzle with servo-motor, the swirler and the ignition device are at high level.

COMBUSTION CHAMBER of a GAS TURBINE INSTALLATION

Fig 20




COMBUSTION CHAMBER ARRANGEMENT

Fig. 21

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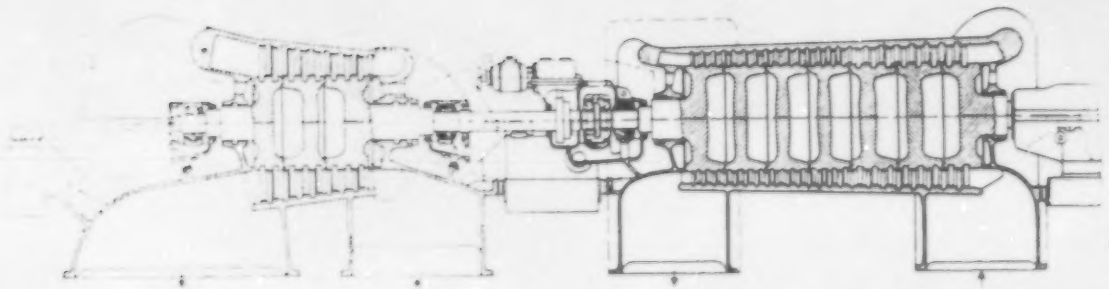
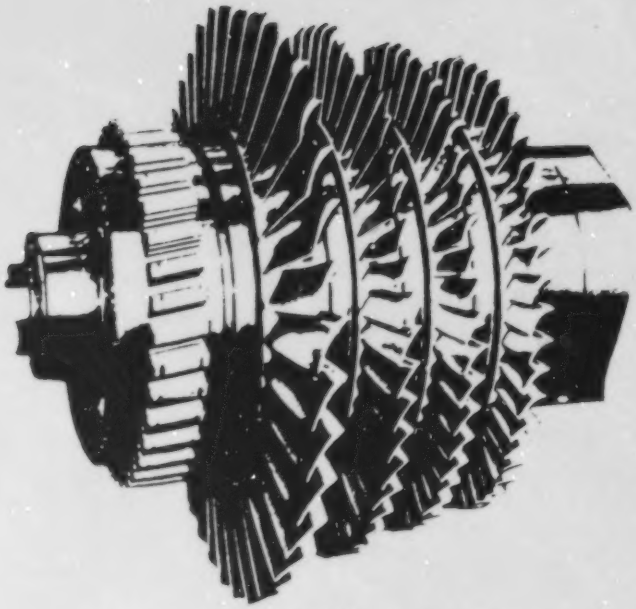


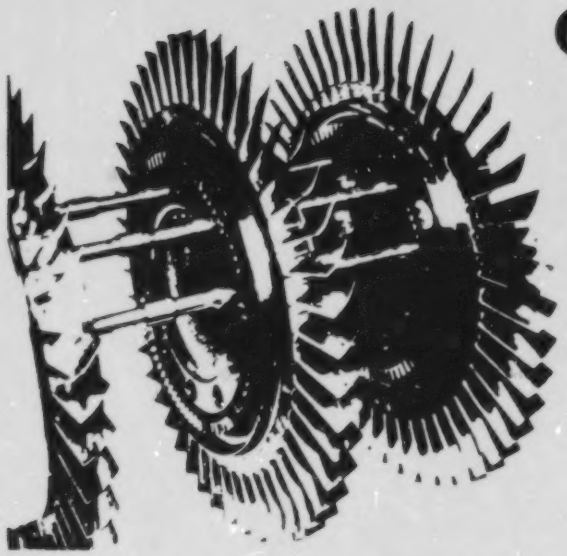
Left, the gas turbine; right, the axial-flow compressor; in between, a thrust bearing with barring gear.
The drawing does not show the jacket at the inlet to the gas turbine and the diffusors at the outlets from the gas turbine and the compressor.

Section through a GAS TURBINE Set

Fig. 22



Last row of blades has not been inserted. (a)



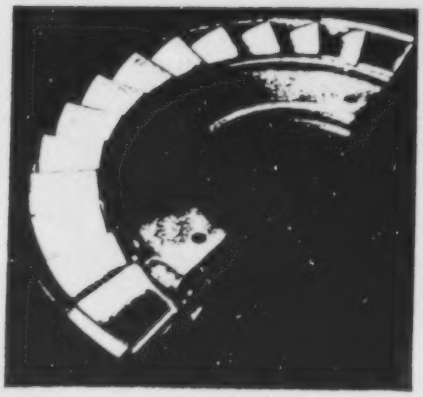
Individual discs are connected by a Curvic coupling and made rigid by a number of through bolts. (b)

GAS TURBINE ROTOR of BOLTED CONSTRUCTION

Fig. 23

(a)

TURBINE MOVING BLADE



(b)

TURBINE FIXED BLADE
(Half Diaphragm)

Fig. 24

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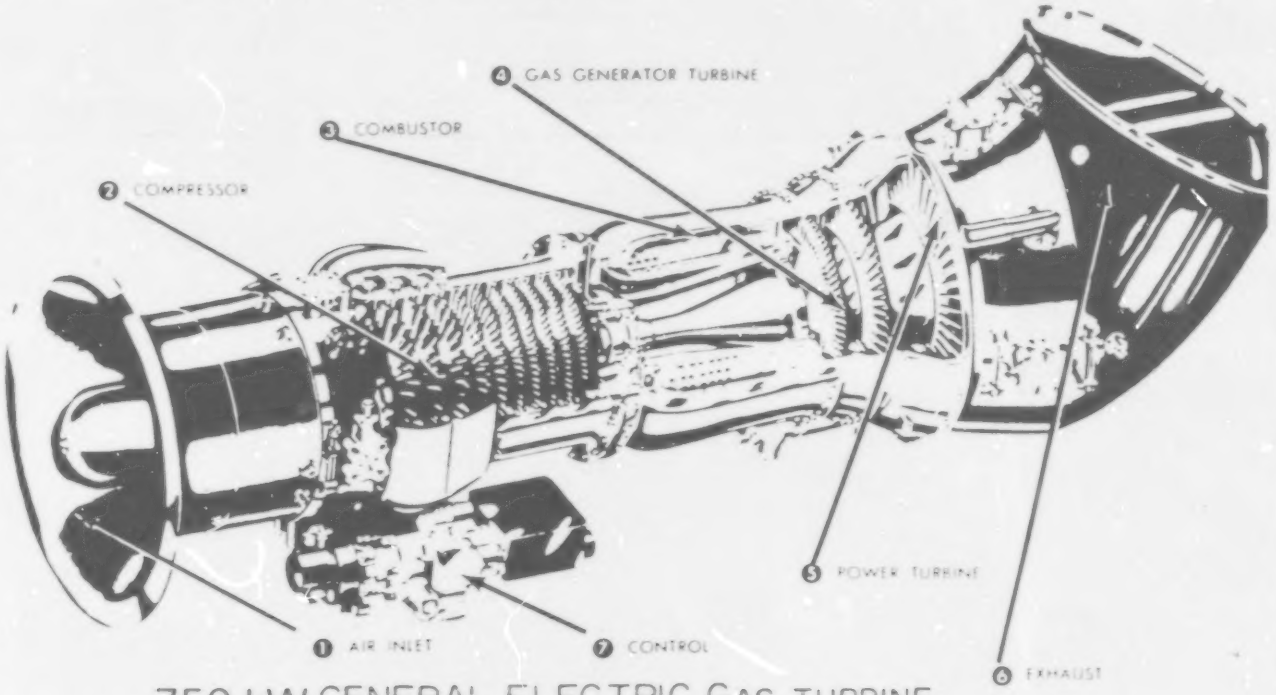
## TURBINE ILLUSTRATIONS

The following illustrations show gas turbines of various types and manufacture.

Fig. 25 shows a General Electric two-shaft machine for use in mechanical drives, rated at 750 kW, thermal efficiency 21%, power turbine shaft speed 19,500 rev/min. In this case the main shaft is split between the two turbines.

The air enters at inlet (1), is compressed in the ten stage axial flow compressor (2) to eight atmospheres, passes to the annular shaped combustor (3), where it is mixed with fuel and ignited by spark plugs which operate only during the starting cycle. The resulting combustion gases first expand through the two stage gas generator turbine (4) which drives the compressor and then through a power turbine (5) which is mechanically independent of the gas generating sections of the engine. The spent gases are then exhausted to atmosphere (6). The control (7) regulates power output by varying fuel flow.




750 kW GENERAL ELECTRIC GAS TURBINE

Fig. 25

Fig. 26 illustrates a General Electric machine rated to deliver 9,500 kW at a thermal efficiency of about 20%. It is a simple open cycle, single shaft machine. The generator (not shown) is driven through gearing at the air compressor shaft end. The combustion chambers are arranged radially around the compressor casing.

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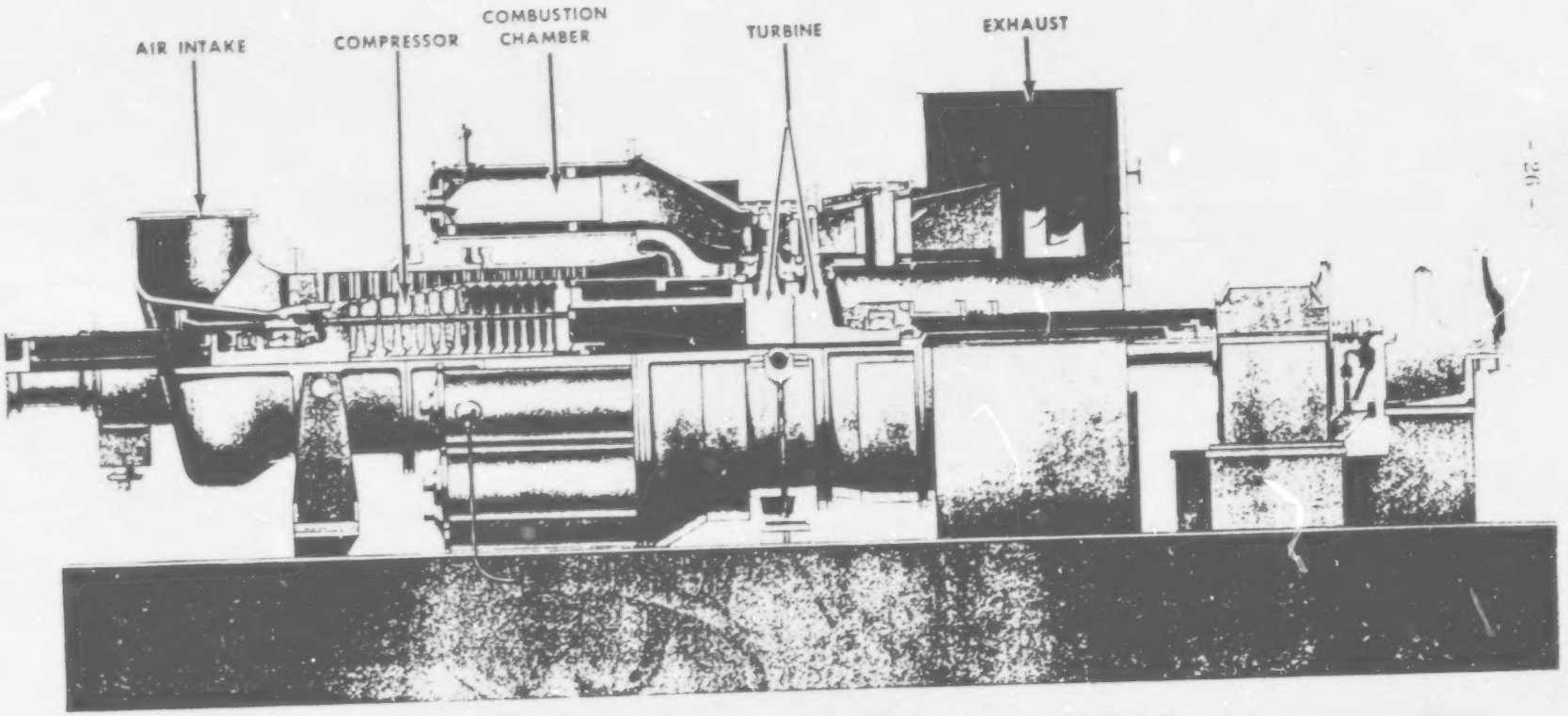
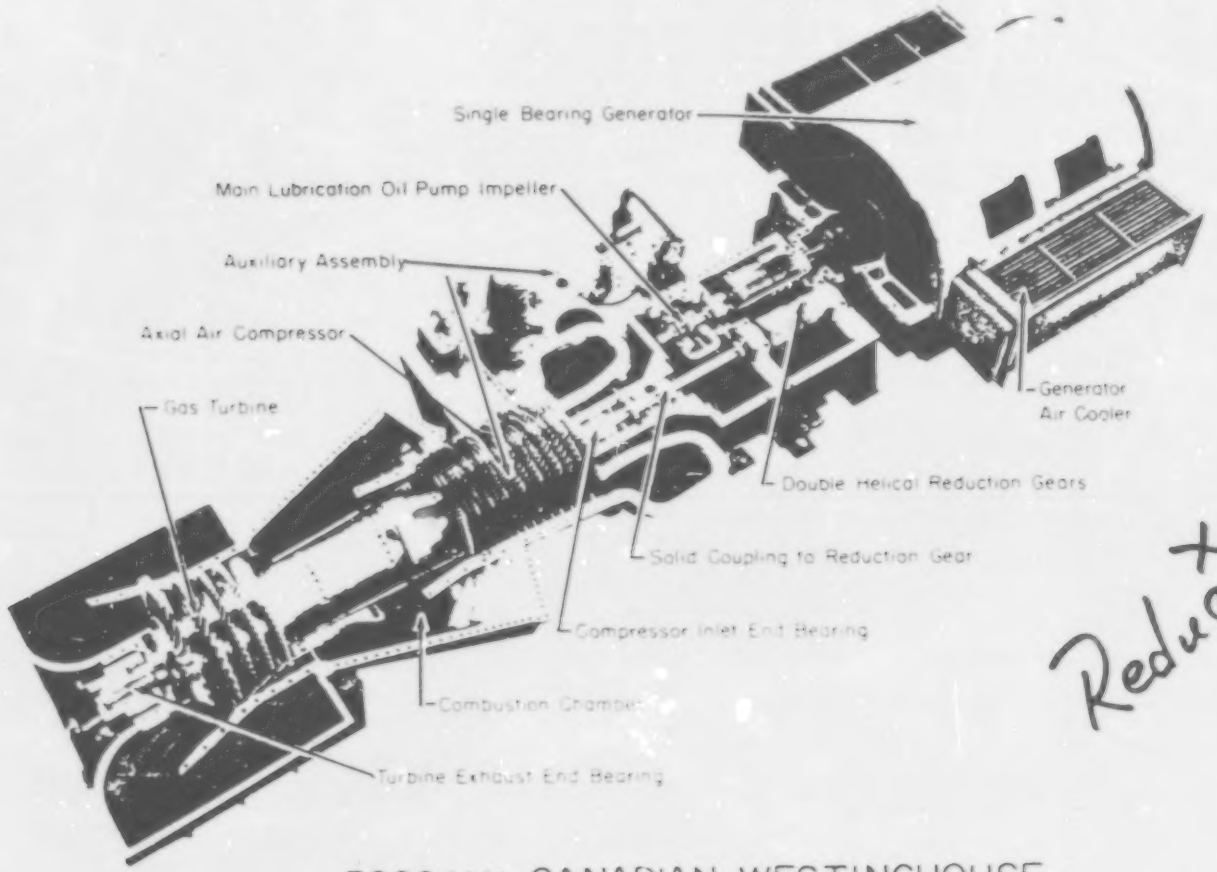



9,500 kW GENERAL ELECTRIC GAS TURBINE

Fig. 2

Fig. 27 shows the layout of a Canadian Westinghouse Co. 5 MW gas turbine plant. It is a simple, open cycle, single shaft machine with six radially arranged combustors.




5000 kW CANADIAN WESTINGHOUSE GAS TURBINE Fig. 27

Fig. 28 is a General Electric Co. dual shaft machine using a regenerative cycle. The air from the compressor discharge is ducted away to the regenerator heat exchanger where it is heated by the exhaust gas from the L-p turbine. It returns to the large, separate combustion chamber and then to the high pressure turbine.

Fig. 29 is an illustration of a Clark Bros. Co. single stage gas turbine rated at 6700 kW with twin, vertical combustion chambers, and a three stage turbine.

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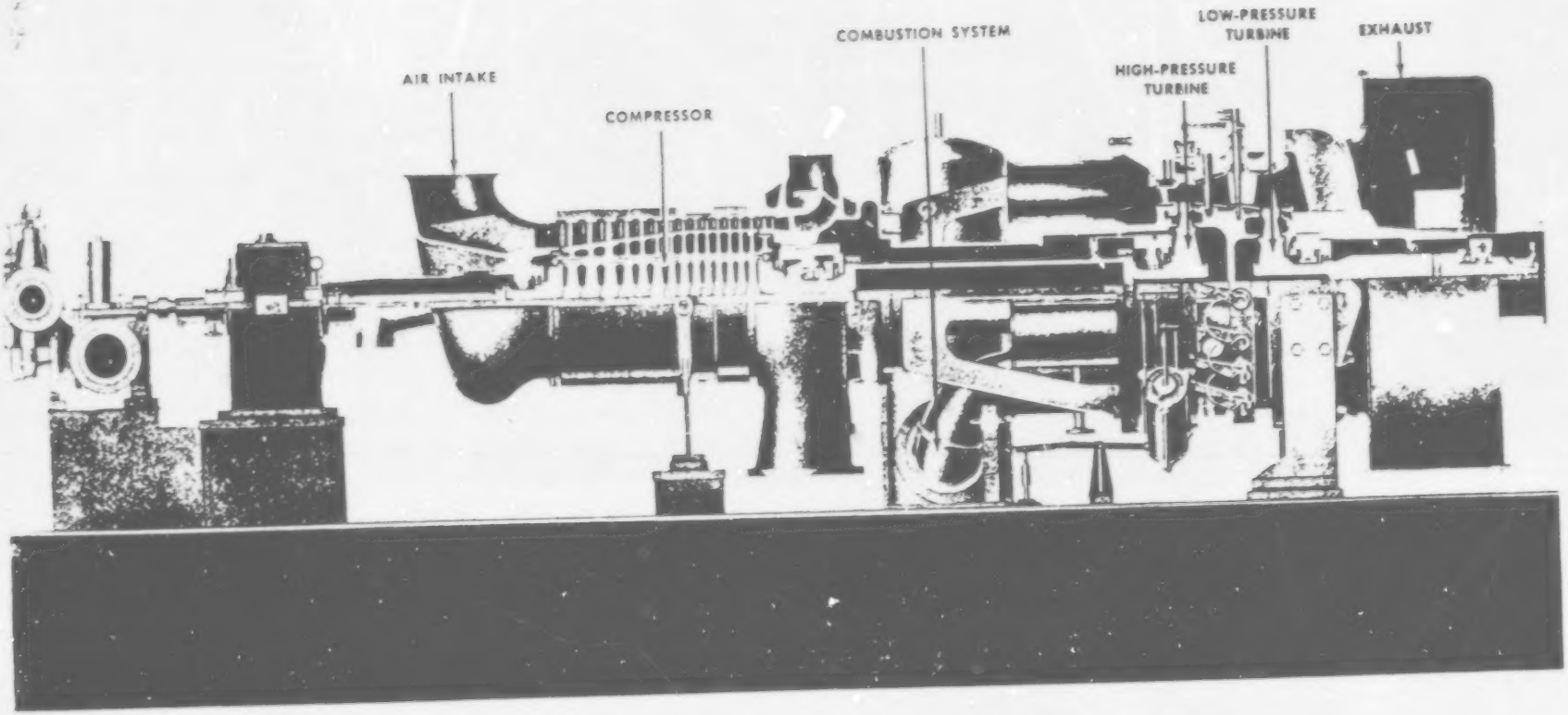
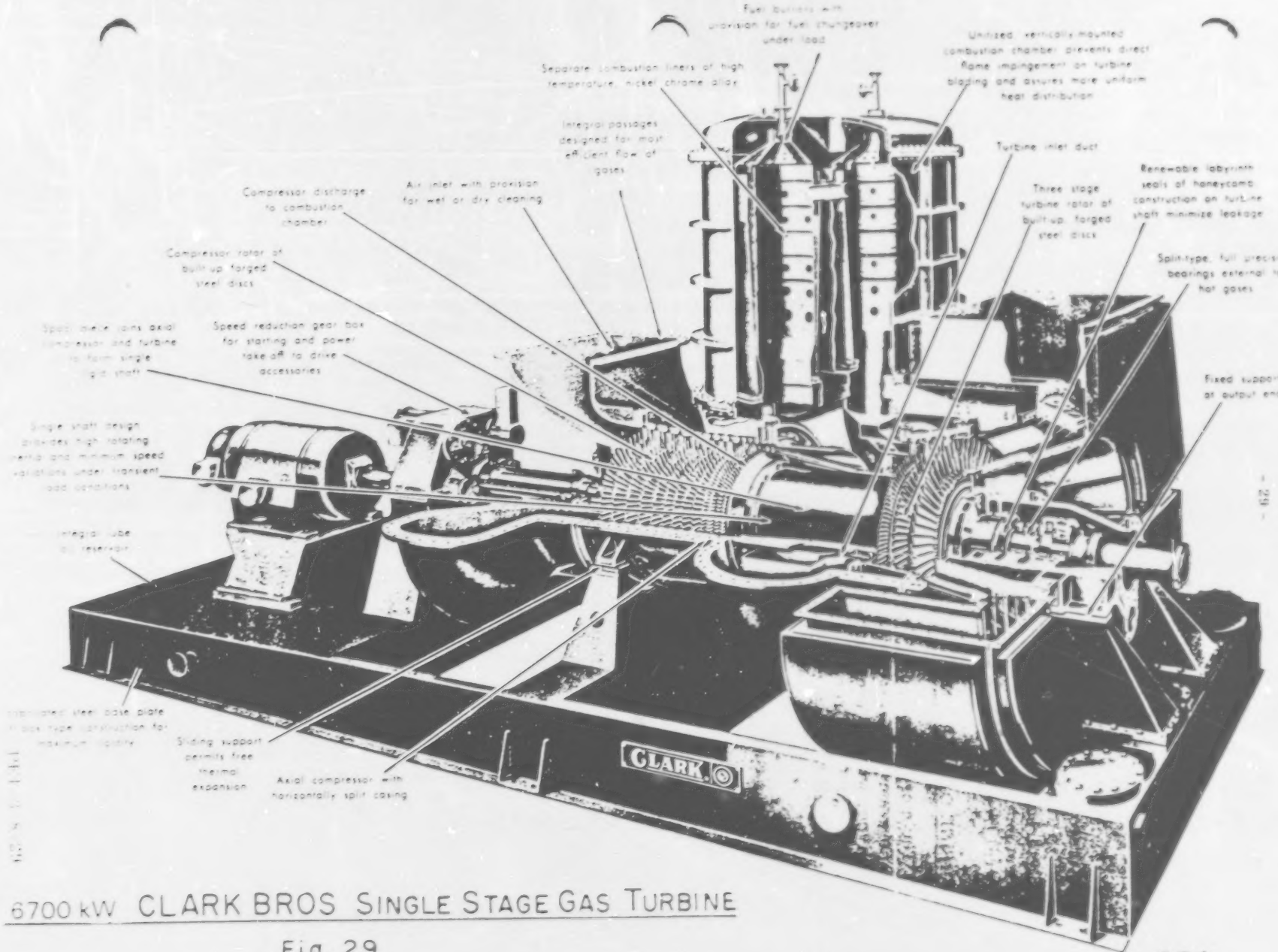



GENERAL ELECTRIC GAS TURBINE with REGENERATOR

Fig 28


Fuel burners with provision for fuel changeover under load
Separate combustion liners of high temperature nickel chrome alloy
Integral passages designed for most efficient flow of gases
Unitized vertically mounted combustion chamber prevents direct flame impingement on turbine blading and assures more uniform heat distribution
Turbine inlet duct
Renewable labyrinth seals of honeycomb construction on turbine shaft minimize leakage
Three stage turbine rotor of built-up forged steel discs
Split-type, full precision bearings external to hot gases
Fixed support at output end
Compressor discharge to combustion chamber
Air inlet with provision for wet or dry cleaning
Compressor rotor of built up forged steel discs
Speed reduction gear box for starting and power take off to drive accessories
Single shaft design provides high rotating inertia and minimum speed variations under transient load conditions
Integral lube oil reservoir
Sliding support permits free thermal expansion
Axial compressor with horizontally split casing
CLARK

6700 kW CLARK BROS SINGLE STAGE GAS TURBINE

Fig. 29





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Fig. 30 shows one of the four 25 MW Brown Boveri gas turbines installed in the Port Mann power station of the B.C. Electric Co. Vancouver. This machine is arranged for intercooling and reheating. There are two shafts, one carrying the L-p turbine, L-p compressor and the generator, and the other carrying the H-p compressor and H-p turbine.

Each shaft is fitted with a starting motor. Combustion is carried out in two large separate combustion chambers, the H-p heating the air at entry to the H-p turbine and the L-p reheating the gas before entry to the L-p turbine. An intercooler is fitted between the L-p and H-p air compressors. The operation of this entire power station is arranged to be carried out by remote control from the load dispatch headquarters of the B.C. Electric Co.

## TURBINE OPERATION AND CONTROL

The most striking difference between steam and gas turbine plants from the operation point-of-view is that in the case of a gas turbine of the open cycle type, with the exception of the fuel control, there are no valves in the main air-gas flow circuit. The pressure differences throughout are small and consequently the pressure loss which would occur through a valve would be excessive.

Thus the quantity of air flowing is dependent only on the compressor speed and this is very nearly constant, especially in the case of a machine used for electricity generation where the compressor is driven on the same shaft as the generator.

The gas turbine output however is dependent, not only upon the quantity of combustion gases flowing, but also upon their energy content, and this allows load control to be carried out by variation of the quantity of fuel burned in the combustion chamber, so that in cases where the machine speed is substantially constant the gas turbine output is directly proportional to the fuel burned.

A single shaft machine driving a generator will be fitted with a speed sensitive governor with a temperature sensitive limit control on the fuel flow. While running up to speed, and at times of part loads, the speed governor will control the fuel flow and the inlet temperature will be below its optimum figure. At full load the temperature control will prevent the turbine inlet temperature from exceeding the safe limit set for the blade material. The output obtainable from the machine will now be dependent on the ambient air temperature.

### Ambient Air Temperature

Gas turbine ratings are usually quoted on the basis of 27$^{o}$C and 100 kPa atmospheric conditions. Variations in the ambient air temperature have a marked effect upon the efficiency and the output; as a rough approximation the output will vary by 1% with 1$^{o}$C change. In view of this some machines have an element included in the governing system to protect the machine against excessive fuel flow which might occur in the case of overload at a time when the ambient temperature is high. This consists of a temperature sensitive element located in the air inlet duct and connected to a fuel valve which will limit the maximum fuel flow according to prevailing ambient temperature.



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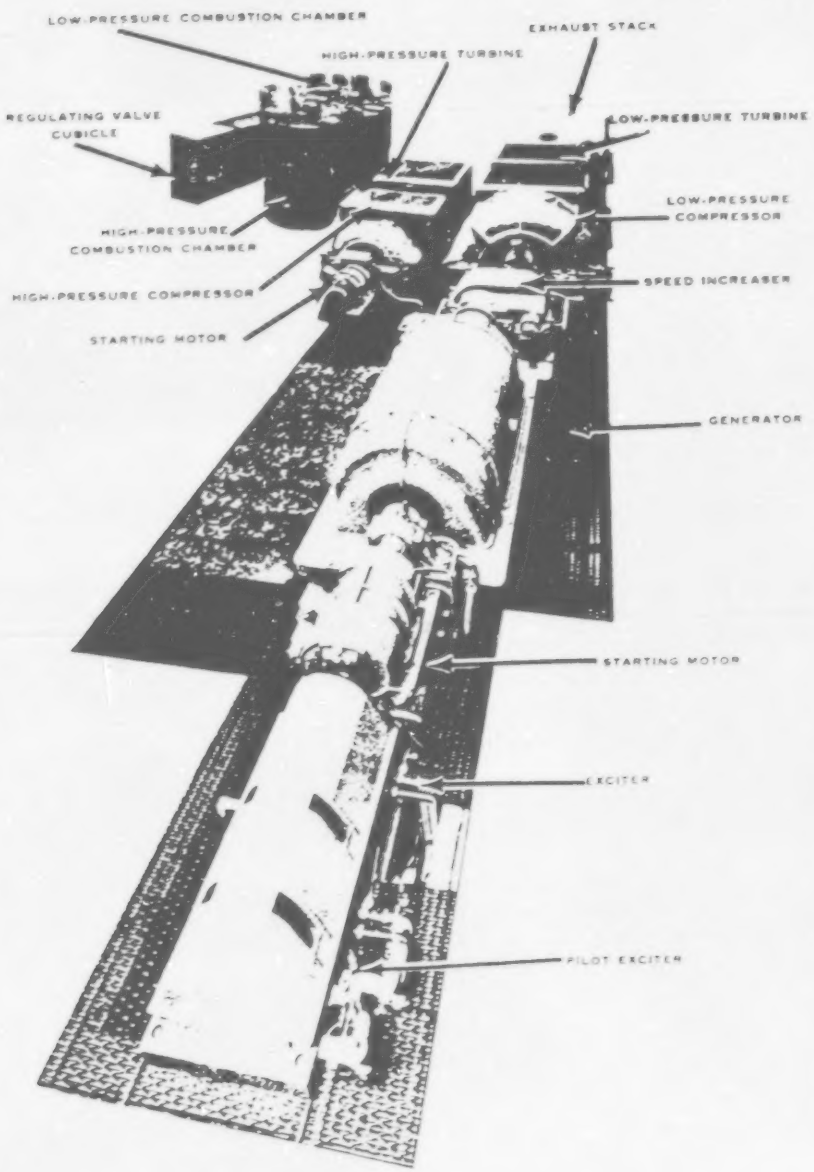


25 MW BROWN BOVERI

GAS TURBINE Fig.30

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Fig 31 shows a diagrammatic arrangement of the fuel system for an A.E.I. gas turbine set of the simple cycle, non-regenerating, single shaft type. Protective devices included are overspeed trip valve, low fuel-gas pressure, low bearing oil pressure, low compressor delivery air pressure.

These protective devices in operation will close the trip valve in the fuel supply line to the combustors. The trip valve is normally kept open by relay oil pressure acting against a closing spring. In the diagram shown, the manufacturer supplies a duplicate or stand-by trip valve together with piping and test apparatus in order that the operation of the trip valve can be tested daily even though the machine is on load.

Fig. 32 illustrates a typical G.E. control diagram for a single shaft gas turbine, showing the layout from the central control panel, through the speed governor and fuel control system to the turbine. Included is a fuel limit relay sensitive to maximum temperature and to rate of increase of temperature.

Fig. 33 is a similar diagram for a G.E. dual shaft gas turbine. Here there are two speed sensitive governors, one to each turbine shaft with a nozzle control system operating between the two turbines, as in Fig. 28.

Dual Shaft Machines

When the output of a gas turbine is controlled by variation of the inlet gas temperature, the thermal efficiency of the machine suffers at times of low load. A two shaft or dual shaft machine has the advantage of being able to control the load output by variation of the speed of the gas generating turbine and compressor while maintaining the inlet gas temperatures to the turbines at their optimum figures.

The type of control, for example, is applied to the Brown Boveri machine of the type shown in Fig. 34 in which the H-p compressor and turbine speed is varied over a range of 3,600 - 4,500 rev/min, while the L-p or load turbine speed remains constant and the turbine inlet temperatures are kept at the optimum 620°C.

In this system a speed sensitive governor on the load (or L-p) turbine controls the fuel supply to the combustor of the gas generating (or H-p) turbine in conjunction with a slower acting constant temperature control. The fuel flow to the load turbine is only controlled by temperature.

Control Systems

The operation of a gas turbine plant will involve the following general functions: (1) Starting and stopping, (2) Speed or load control, (3) Temperature and rate of temperature change control, and (4) Emergency shut-down control.

The control system employed may be manual, semi-automatic, fully automatic or even a fully automatic arrangement with a remote control. The starting sequence will include energizing of the auxiliaries, engaging the clutch between the starting motor and the set, control of the acceleration of the starting motor and control of the fuel increases required to bring the machine up to operating speed.



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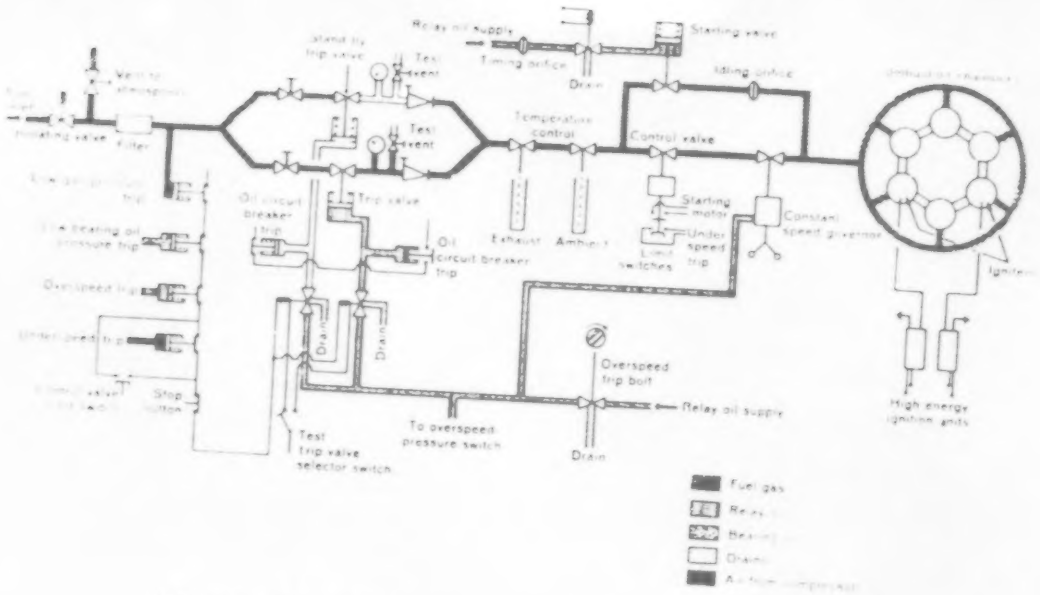
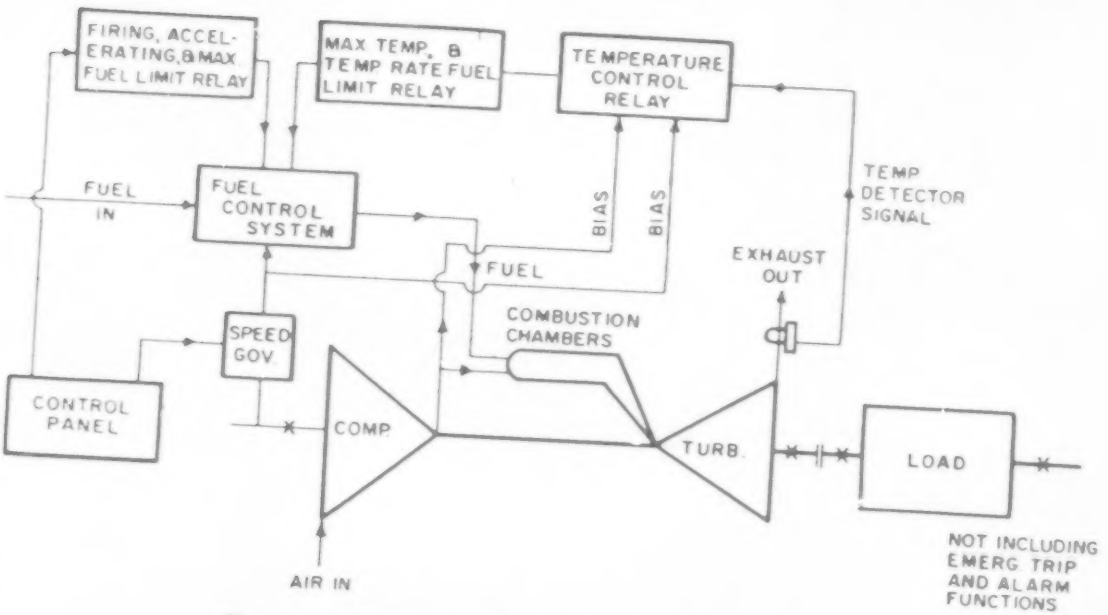



FUEL SYSTEM for A.E.I. GAS TURBINE

-DIAGRAMMATIC ARRANGEMENT- Fig.31




Typical Functional Control Diagram for

SINGLE SHAFT GAS TURBINE Fig.32

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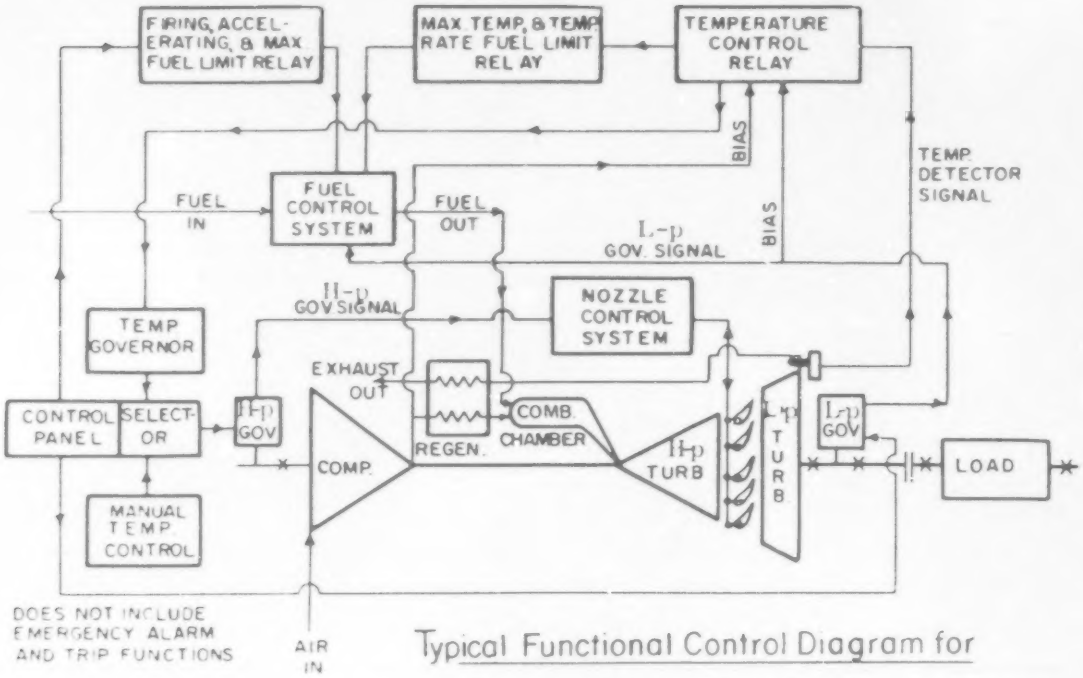
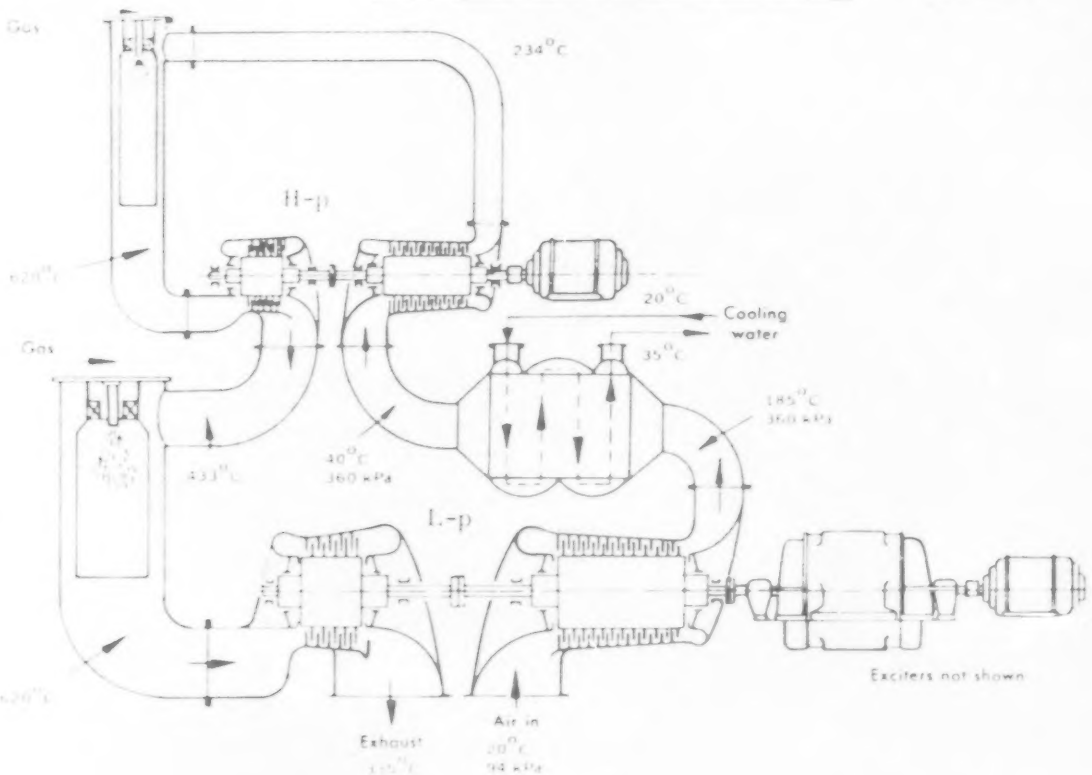



Typical Functional Control Diagram for

DUAL SHAFT GAS TURBINE Fig.33




Typical System

TEMPERATURES and PRESSURES Fig.34

## GAS TURBINE PERFORMANCE CALCULATIONS

The following terms and values are used when quoting the performance of gas turbines.

1. The Work Ratio

This is the ratio of the net or useful output work delivered by the machine, to the work developed by the turbine.

$$\text{Work Ratio} = \frac{\text{Turbine output work - Work input to compressor}}{\text{Turbine output work}}$$

This will be of the order of 40%, in other words, the compressor takes about 60% of the total work developed in the turbine.

2. The Air Rate

This is the mass of air required to produce one kWh. It is expressed as the ratio of the heat equivalent of kWh in kilojoules (3600) to the net work delivered to the shaft per kg of air (kJ).

3. The Air Fuel Ratio

This is the mass of air used per kg of fuel. It is the ratio of the lower heating value of the fuel in kJ/kg to the heat supplied by the combustor in kJ/kg.

4. The Fuel Rate - or specific fuel consumption

Defined as the mass of fuel in kg required to produce kWh.

It will be equal to $\frac{\text{Air Rate}}{\text{Air Fuel Ratio}}$

5. The following are definitions of some terms used:

Compressor Pressure Ratio - the absolute pressure at the compressor discharge divided by the absolute pressure at the compressor inlet.

Machine Efficiencies - refers to the compression efficiency of the compressor and the engine efficiency of the turbine. That is, compression efficiency is the ratio of the work required for ideal compression through a given pressure range, to the actual work required by the compressor.

Turbo-engine efficiency is similar to the 'efficiency ratio' of a steam turbine and is the actual work developed by the turbine in expanding the gas over the ideal work available for the expansion.

Example

A gas turbine working on the air standard Brayton cycle takes in 1150 $m^3$/min, of air at 100 kPa, and 4.5°C. The compressor compression ratio is 5. Fuel burned in the combustor raises the gas temperature of 700°C at the turbine inlet, the turbine exhaust pressure is 100 kPa. If the turbine engine efficiency is 85% and the compression efficiency is 83%, find the net output power and the thermal efficiency of the plant. (Neglect the pressure drop between compressor outlet and turbine inlet.)

Solution

The method used can be summarized as follows:

Calculate work done per kg on the air by the compressor and the work done per kg by the gas in the turbine. The difference will give the net output work per kg. Find the kg of air flowing per min, then the net output power

$$= \frac{\text{kg air/min} \times \text{net work done (kJ/kg}}{60} = \text{kW}$$

and the thermal efficiency

$$= \frac{\text{Output work}}{\text{Heat supplied}}$$

These results can be obtained by two methods:

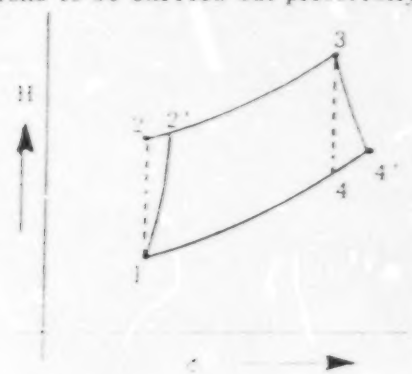
(a) By using basic gas laws and calculating the various gas conditions at each stage in the cycle; or

(b) By the use of Gas Tables

In (a), the specific heats of the gas must be assumed to be constant. Cp will be taken as being 1.005 and $\gamma$ as 1.4 for this example. It will be assumed that air is the working medium throughout, characteristic constant R = 0.2871 kJ/kg K.

These assumptions introduce errors and furthermore the calculations can become complicated. The Gas Tables (b) take into account the variations in specific heat and their readings are consequently more accurate; their use also simplifies the working of such problems.

An Enthalpy and Entropy chart for air (similar to the Mollier Chart for steam) is available from information extracted from the Gas Tables, and this allows calculations to be carried out pictorially as was done for steam turbines.




The cycle would be as represented on the HΦ diagram, Fig. 35, point 1 representing compressor intake conditions. Point 2 will be the point reached after ideal or isentropic compression and point 2' that reached after the actual compression.

Point 3 represents the conditions at inlet to the gas turbine. Point 4 would be the point reached after an ideal expansion and point 4' that of the actual expansion.

Fig. 35

PE1-3-8-36

Example

Given $P_1 = 100$ kPa $\quad T_1 = 4.5 + 273 = 277.5$ K

$T_3 = 700 + 273 = 973$ K

$P_4 = 100$ kPa

Compression Ratio $= 5 = \frac{V_1}{V_2}$

Assume Cp for air constant at 1.005

$\gamma = 1.4$ and R for air $= 0.2871$ kJ/kg K

To find $P_2$

$$P_1 V_1^{\gamma} = P_2 V_2^{\gamma}$$

$$\therefore P_2 = P_1 \times \left(\frac{V_1}{V_2}\right)^{\gamma} = 100 \times 5^{1.4} = 100 \times 9.518$$

$$= \underline{\underline{951.8 \text{ kPa}}}$$

This is the compressor discharge pressure. Assuming no pressure drop between compressor and turbine then $P_3$ is also 951.8 kPa i.e. $P_2 = P_3$

Note the pressure ratio of the compressor

$$\frac{P_2}{P_1} = \frac{951.8}{100} = \underline{9.518}$$

To find $V_1$ and $V_2$

$$P_1 V_1 = w R T_1$$

$$V_1 = \frac{w R T_1}{P_1} = \frac{1 \times 0.2871 \times 277.5}{100} = 0.7967 \text{ m}^3$$

and $\frac{V_1}{V_2} = 5 \quad \therefore V_2 = \frac{0.7967}{5} = 0.1593 \text{ m}^3$

PE1-3[illegible]-37



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To find $T_2$

$$P_2V_2 = w R T_2$$

$$T_2 = \frac{P_2V_2}{w R} = \frac{951.8 \times 0.1593}{1 \times 0.2871} = 528 \text{ K}$$

To find $T_4$

$$\frac{T_4}{T_3} = \left(\frac{P_4}{P_3}\right)^{\frac{\gamma-1}{\gamma}}$$

$$T_4 = 973 \left(\frac{100}{951.8}\right)^{\frac{1.4-1}{1.4}} = 973 (0.1051)^{0.286}$$

$$= 973 \times 0.525$$

$$= 511 \text{ K}$$

$$\text{Compression Efficiency} = \frac{\text{Ideal Temp. Change}}{\text{Actual Temp. Change}}$$

$$\therefore 0.83 = \frac{T_2 - T_1}{T_2^1 - T_1} = \frac{528 - 277.5}{T_2^1 - 277.5}$$

$$T_2^1 = \frac{(528 - 277.5)}{0.83} + 2775.5 = 301.8 + 277.5$$

$$= \underline{579\ K}$$

$$\text{Turbine Efficiency} = \frac{\text{Actual Temp. Change}}{\text{Ideal Temp. Change}}$$

$$\therefore 0.85 = \frac{T_3 - T_4^1}{T_3 - T_4} = \frac{973 - T_4^1}{973 - 511}$$

$$T_4^1 = 973 - 0.85\ (973 - 511) = 973 - 393$$

$$= \underline{580\ K}$$

Work done by Compressor/kg of air (Ideal)

$$= Cp\ (T_2 - T_1)$$

$$= 1.005\ (528 - 277.5) = 264\ kJ/kg$$

$$\text{Actual work done/kg} = \frac{264}{0.83} = \underline{\underline{318.4\ kJ/kg}}$$

Work done by turbine/kg of gas

$$= Cp\ (T_3 - T_4)$$

$$= 1.005\ (973 - 511) = \underline{464.3\ kJ/kg}$$

Actual work done/kg = $464.3 \times 0.85$ = <u>394.7 kJ/kg</u>

Net work output/kg (Ideal) = $464.3 - 264$ = <u>200.3 kJ/kg</u>

Net work output/kg (Actual) = $394.7 - 318.4$ <u>76.26 kJ/kg</u>

PE1-3-8-39



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Volume of air flowing/minute = $1150\ m^3$

To find mass of air kg/min

$$P_1V_1 = w\,R\,T_1$$

where $P_1 = 100$ kPa, $V_1 = 1150\ m^3$, $R = 0.2871$, $T_1 = 277.5$ K

$$w = \frac{P_1V_1}{R\,T_1} = \frac{100 \times 1150}{0.2871 \times 277.5}$$

$$= 1443.4 \text{ kg/min}$$

Power output (Ideal) $= \frac{200.3 \times 1443.4}{60} =$ **4818.5 kW**

Power output (Actual) $= \frac{76.26 \times 1443.4}{60} =$ **1834.6 kW**

Heat supplied to the cycle/kg air (Ideal)

$= Cp\,(T_3 - T_2)$

$= 1.005\ (973 - 528)$

$=$ **447 kJ/kg**

Heat supplied to the cycle/kg air (Actual)

$= Cp\,(T_3 - T_2{}^1)$

$= 1.005\ (973 - 579)$

$=$ **396 kJ/kg**

Thermal efficiency (Ideal) $= \frac{\text{Work Done}}{\text{Heat Supplied}} = \frac{200.3}{447}$

$=$ **44.81%**

Thermal efficiency (Actual) $= \frac{76.26}{396} =$ **19.26%**

Note the marked effect of the turbine and compressor efficiencies upon the thermal efficiency of the plant.

This illustrates the absolute need for high efficiency compression in particular. Reduction in this efficiency would mean that the point would soon be reached at which the turbine would be incapable of producing enough work to drive the compressor and the machine would not operate at all.

## GENERAL INSTRUCTIONS
### for Starting and Stopping

As in the case of steam turbines the manufacturers of any particular gas turbine set will give precise instructions for its operation and these should be followed very closely. The following can be taken as general remarks on the subject.

### Starting

Check the lubricating oil tank for correct level and start the auxiliary oil pump. See that this is delivering oil to all bearings.

Check the cooling system and start the circulating water pumps. This will supply oil coolers, generator air coolers, and compressor intercoolers, where these are fitted.

Check that the fuel supply valves to the combustors are closed off.

The larger machines will be fitted with jacking oil pumps; in this case start this pump in order to ease the mass of the shafts in the bearings.

Start the barring motor and then the starting motor. Check the starting current and speed as this motor accelerates the turbine shaft up to the recommended speed. This must be sufficient to enable the air compressor to supply air to the combustors and will be upwards of 20% of the full speed.

As a rough guide it can be taken that a gas turbine set requires a rotation of about 50% of the full rated speed to obtain a sufficiently high air pressure to be self sustaining.

The starting motor power required to attain this speed in 1 minute will be about 5% of the rated output of the set.

Supply ignition to the combustors and begin admitting fuel. See that ignition is satisfactory. Increase fuel and when combustion is stable switch off the ignition.

The temperature at inlet to the turbine must be carefully watched as the turbine speed increases, and kept within the recommended limit.

The starting motor is generally arranged to cut out automatically by a centrifugal clutch or other means.

The main lubricating oil pump and the speed governor will be gear-driven from the main shaft and will come into operation as the speed builds up. Within about 10% of normal running speed, the governor will take control. The main oil pump will indicate full running condition when its oil discharge pressure reaches a steady figure.

The total time taken in running up from standstill will be about 10 minutes. Adjust the machine speed to correspond with the system frequency with which the generator is to be paralleled, synchronize and close main switch.

Raise load as required. A gas turbine can be loaded rapidly since the expansion problems are not so acute as a steam turbine. The time taken from no load to full load on a 30 MW machine is quoted by Brown Boveri as being about 10 minutes.

Shutting Down

Reduce the load on the generator.

Open main switch.

Leave the machine idling for a few minutes.

Start the auxiliary oil pump and see that it is delivering oil satisfactorily.

Change over fuel supply to hand control and shut off fuel.

See that ignition ceases in the combustor and allow the machine to come to rest.

During this time control the circulating water supplies to give the required lubricating oil temperatures.

After the machine has stopped, run the jacking oil pump and then start the barring motor to keep the shafts rotating slowly during the cooling down period.

Notes on Operating

Whilst in operation the gas turbine will require supervision to ensure that lubricating oil flow and temperatures are satisfactory, cooling water flow is adequate to maintain oil cooler and generator air cooler conditions, correct operation of inlet-gas temperature regulator etc. Some machines, for example, Brown Boveri are provided with sight glasses into the combustors to observe the combustion conditions, and into the turbine inlet stage to observe the appearance of the inlet stationary diaphragm blading and the first row of moving blading.

Safety devices, such as the machine over-speed trip, should be tested at regular intervals. Sequential interlocks are often provided to protect the machine against operation of starting motor and ignition in incorrect sequence. Ignition should not be switched on when the machine is standing since there may have been some fuel leakage into the combustor and there is danger of explosion under these conditions. Running the compressor by the starting motor will ensure sufficient air flow to dispel any residual fuel gases.

## QUESTIONS

POWER ENGINEERING

First Class
Sect. 3, Lect. 8

1. In what circumstances would you expect to find a gas turbine plant chosen as the prime mover for power production or for electricity generation?

2. Explain with the aid of a diagram the layout and operation of a simple, open cycle gas turbine.

3. Review the possible improvements to this cycle.

4. Why is a starter motor necessary for a gas turbine.

5. What is meant by a closed cycle as applied to a gas turbine. Give the advantages and disadvantages.

6. What is a Free Piston gas generator, how does it operate?

7. Give the types of air compressors used with gas turbines. Why are these types chosen?

8. What are the fuels used for gas turbines? Can coal be used for this purpose?

9. State the main differences between steam and gas turbines.

10. How is load control carried out on a gas turbine used for electricity generation?